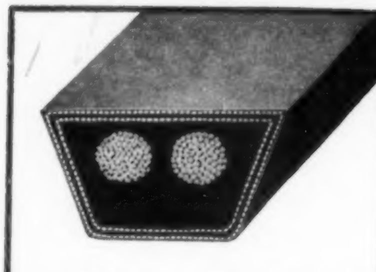
MACHINE DESIGN

July 1953

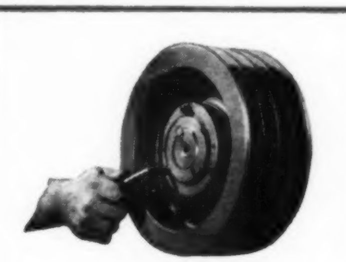
PRECISION FORGINGS

Page 3



TEXROPE Grommet V-Belts—20 to 50% longer life than ordinary belts. Size for size, grommet belts give 1/3 more gripping power, pull heavier loads with higher safety factor.

STANDARD CAST IRON SHEAVES — Made to order from stock patterns. One to 14 grooves for A, B, C, D or E belts. One to 1000 horsepower. Split sheave or split hub.



MAGIC-GRIP SHEAVES — Semi-steel casting with interchangeable, patented taper bushings. On and off in a jiffy. Cannot shake loose. Two to 12 grooves for B, C and D belts; 2 to 250 hp.

Everything in V-Belt Drives

from one reliable source!

ALLIS-CHALMERS HAS BOTH the engineering experience and the drive equipment to lick the most unusual problems. Your special problems are carefully analyzed by a staff of experienced application engineers familiar with your industry. The result — the right drive for the job! Call your nearby Allis-Chalmers distributor or district office for help on your drive problems, or write Allis-Chalmers, Milwaukee 1, Wisconsin. A-4087

Texrope and *Magic-Grip* are Allis-Chalmers trademarks.

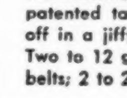
ALLIS-CHALMERS

Sold . . .

Applied . . .

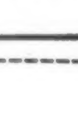
Serviced . . .

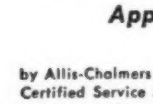
by Allis-Chalmers Authorized Distributors, Certified Service Shops and Sales Offices throughout the country.



MOTORS — ½ to 25,000 hp and up. All types.

CONTROL — Manual, magnetic and combination starters; push button stations and components for complete control systems.





PUMPS — Integral motor and coupled types from ¾ in. to 72 in. discharge and up.

THE PROFESSIONAL JOURNAL FOR ENGINEERS AND DESIGNERS

MACHINE DESIGN

July, 1953
Vol. 25—No. 7

REGULAR DEPARTMENTS

USE POSTAGE-FREE CARDS FOR:

Over the Board

The Thieving Transformer

We've heard engineers abuse balky equipment and even cast aspersions on the ancestry and moral habits of machine components. The Soviets as usual are 'way ahead of us and are now introducing politics into their descriptions of equipment. According to a book published in Yugoslavia and quoted in the *General Electric Review,* there are two types of transformers: *der kapitalistische Raubtransformator* (the capitalistic robber-transformer) and the new socialistic transformer. Needless to say the former is an inefficient, decadent device while the latter is a paragon of efficiency. However the *G. E. Review* reveals, by direct quotation from Russian engineering publications, that Soviet engineering falls far short of what some popular writers would have us believe.

This Month's Cover

That out-size red arrow you see on the front cover represents the squeeze required to produce an extrusion forging. It is part of artist George Farnsworth's idea of the wedding of impact extrusion and press forging. Termed "press extruded forging," the offspring has attractive possibilities for cost-conscious designers, which are discussed in Christensen's article beginning on Page 124.

Incidentally, our fresh approach to front cover design has been recognized by the award of a certificate of merit for the "best graphical presentation" in the annual *Industrial Marketing* editorial achievement competition. For more news of honors won by MACHINE DESIGN in this competition, turn to Page 183.

Design Details

Starting on Page 152, a new department makes its initial appearance in MACHINE DESIGN. Entitled Design Details, it has been created specifically to assist the designer in selecting and applying standard commercial parts. Through these pages, we plan to present a pictorial guide of practical and helpful suggestions on how to solve those "little" problems which can make such a big difference in design. As a forerunner of things to come, the first of this series is devoted to retaining rings—a useful and versatile fastener for machine assemblies. We hope you will find some cost-saving hints in this and the installments to follow.

For Your Convenience

Beginning this month you will find some rearrangement of certain sections of MACHINE DESIGN following the main feature articles. Engineering News Roundup is now in the position formerly occupied by Design Abstracts which has been moved to a position where the pages can run consecutively without the long "breakover" formerly necessary. New Parts and Materials now follows the Helpful Literature insert instead of preceding it; other changes of a minor nature also have been made, all designed to improve readability. If you have any difficulty finding your favorite department, just turn back one page to the table of contents.

EDITORIAL STAFF

COLIN CARMICHAEL
Editor

ROGER W. BOLZ, *Associate Editor*
BENJAMIN L. HUMMEL, *Associate Editor*
ROBERT L. STEDFELD, *Associate Editor*
LEO F. SPECTOR, *Assistant Editor*
KEITH A. CARLSON, *Assistant Editor*
ROBERT C. RODGERS, *Assistant Editor*
JANE H. SMITH, *Assistant Editor*
FRANK H. BURGESS, *Art Editor*

New York .... B. K. PRICE, L. E. BROWNE
SAMUEL W. BAKER
Chicago .... ERLE F. ROSS
Pittsburgh .... ROBERT E. HALL
Detroit .... FLOYD G. LAWRENCE
Washington .... E. C. KREUTZBERG
London .... VINCENT DELPORT

BUSINESS STAFF

ROBERT L. HARTFORD
Business Manager

MARY L. CALLAHAN, *Service Manager*
RICHARD J. STRNAD, *Production Manager*
JACK C. GERNHARD, *Circulation Manager*

Chicago .... HOWARD H. DREYER
RICHARD K. LOTZ
New York .... RUSSELL H. SMITH
EDWARD S. LAWSON
ALAN C. BUGBEE
Cleveland .... HAROLD B. VEITH
JACK W. WALTON
Los Angeles .... F. J. FULLER
Griffin, Ga. .... FRED J. ALLEN

MAIN OFFICE

Penton Building, Cleveland 13, Ohio
Main 1-8260

Branch Offices

New York 17 .... 60 East 42nd St.
Murray Hill 2-2581
Chicago 11 .... 520 North Michigan Ave.
Whitehall 4-1234
Pittsburgh 19 .... 2837 Koppers Building
Atlantic 1-3211
Detroit 2 .... 6560 Cass Ave.
Trinity 5-3024
Washington 4 .... 1123 National Press Bldg.
Executive 6849
Los Angeles 48 6262 Commodore Sloat Drive
Webster 1-6865
Griffin, Ga. .... 331 South Twelfth St.
London, S.W.1 ... 2 Caxton St., Westminster

Published by
THE PENTON PUBLISHING COMPANY

G. O. HAYS .... *President and Treasurer*
E. L. SHANER .... *Chairman*
R. C. JAENKE ... *Vice Pres., Director of Adv.*
F. G. STEINEBACH ... *Vice Pres. and Secretary*
J. P. LIPKA ...... *Asst. Secy. and Asst. Treas.*

Also publisher of
Steel • Foundry • New Equipment Digest

Published on the seventh of each month. Subscription in the United States and possessions, Canada, Cuba, Mexico, Central and South America: One year $10. Single copies, $1.00. Other countries one year, $15.

Acceptance under Section 34.64 P. L. and R. authorized.

INDEX

Advertising and editorial content itemized for convenience when studying specific design problems

(*Concluded on Page* 10)

INDEX *(Concluded from Page 7)*

MACHINE DESIGN is indexed in Industrial Arts and Engineering Index Service, both available in libraries, generally

MACHINE DESIGN
JULY 1953

Only Total Achievement Counts

THE Old Timer was adamant. Nothing but a Jackrabbit starter would do. For fifty years he, and his father before him, had been specifying Jackrabbits and "they don't come any better." Against their own better judgment his more alert colleagues allowed the Old Timer his way, but what happened?

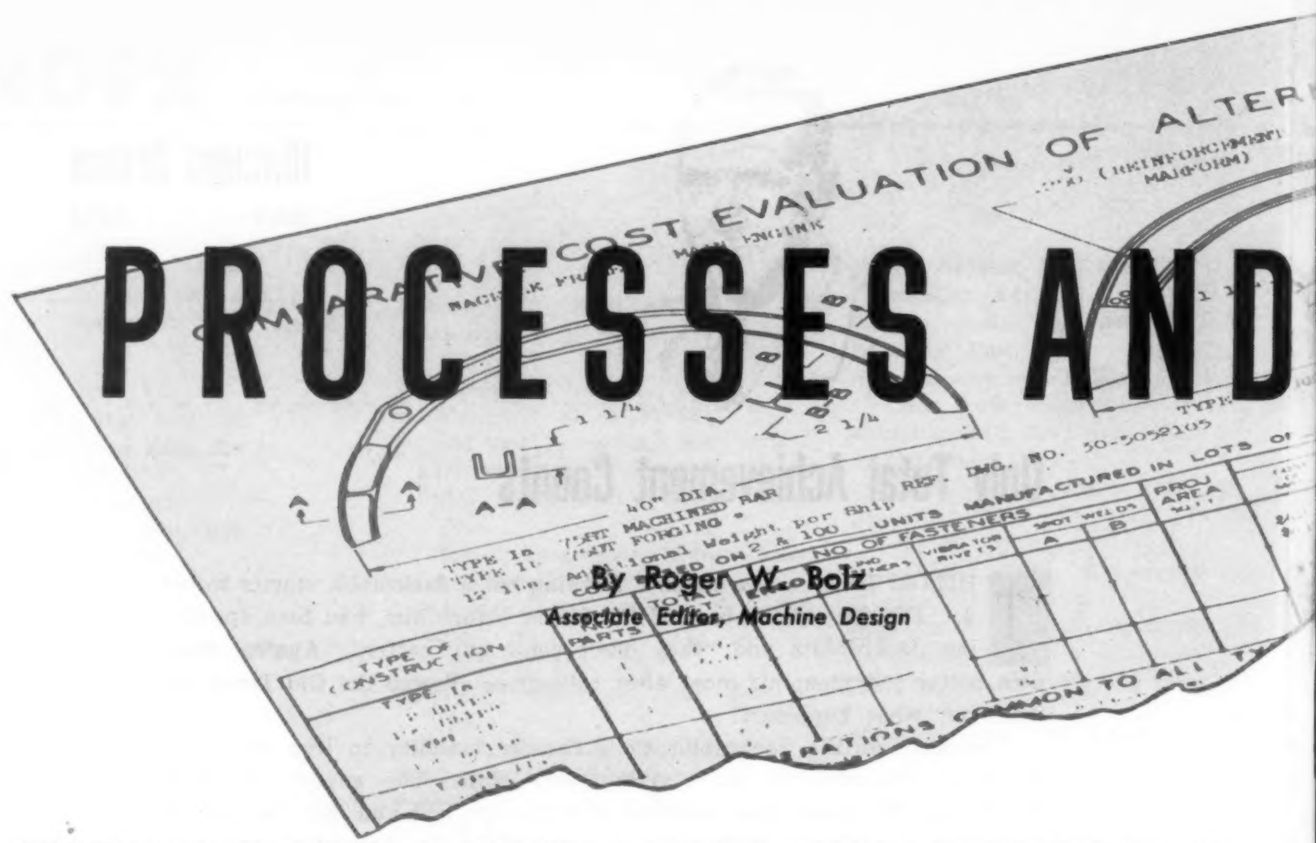
Because of the Jackrabbit Co.'s chronic inability to iron out labor disputes, deliveries of the starters were slow. The superb machines which the Old Timer had designed stood incomplete and idle for lack of one vital component, production departments were disrupted, and customers squawked. As the starters began dribbling in, the machines were brought back to the assembly floor for completion and things returned to normal; but not before thousands of dollars were wasted in lost motion to say nothing of vanishing customer goodwill.

What went wrong? In the first place the Old Timer had failed to keep abreast of new developments in machine components. Otherwise he would have known that competitors of the Jackrabbit were providing equally good, often better, features at a lower price. Again if he had consulted his purchasing agent he would have known that Jackrabbit was notoriously unpredictable on deliveries. With the facts in his possession he wouldn't have been so stubborn.

The moral is obvious. The percentage of the cost of a new machine that goes into purchased items is so high that selection and specification of components deserve as careful analysis as calculation of stresses and design for producibility. It is short-sighted, at the very least, to freeze a design around a particular proprietary item without a thoroughgoing study of available alternates and a check on such elementary matters as delivery and price, which are part and parcel of today's engineering design picture.

The Old Timer was a technical success but in terms of overall engineering accomplishment he was a dismal failure. He ignored the vital principle that enough of the right material must be in the right place at the right time if his design is to become a reality. Only total achievement counts.

Colin Carmichael
EDITOR



PROCESSES AND

By Roger W. Bolz
Associate Editor, Machine Design

ALTHOUGH the design engineer, to be eminently successful, must be original and visionary, he must also produce final designs which are practical. Difficult or expensive manufacturing characteristics today must be classed as equivalent to functional mediocrity. To be eminently practical the designer must create products capable of manufacture by processes incurring lowest possible costs. Competent analyses indicate that few, if any, designs are created that cannot be reduced to an assembly of parts producible by the more commonly available methods.

Design Influence: Practically speaking, as part designs are created, developed and detailed, the designer is actually selecting the process or processes by which the parts will be manufactured. Or, too, he may be limiting the useable methods to those least desirable. Drawing details and specifications actually place wide limits or narrow restrictions on practical processing. The design of the product often determines whether the shop can be set up to manufacture with relatively few operations or will be strapped with costly, unpredictable or even unnecessary steps in production.

As an example, operations such as hand filing, burring, polishing or snagging are unpredictable and often defy all time standards. Not only lower costs but improved parts are the result when these and similar undesirable operations are avoided.

From these and other pertinent considerations, it can be said with assurance that final costs are dictated by the design engineer. The specifications set down on the engineering drawings for the parts of any particular machine place firm limits on production costs which seldom can be altered to any significant degree without basic redesign. This fact makes it imperative that the design engineer know and recognize his influence on production costs; he can achieve amazing cost reductions with a well-rounded knowledge of how parts can best be produced, and how production costs are affected by quantity, process and tolerance.

While numerous production innovations are generally adopted to attain acceptable manufacturing costs, in the overall analysis the greatest achievements in the direction of economy are most easily attained through proper design. Regardless of type of machine parts, quantities or tolerances involved, maximum quality and minimum costs can be obtained only if due consideration is given the possible production methods available for their manufacture.

It may be argued that methods, planning and cost analysis are not the responsibility of the designer and, considered as specific on-the-job duties, this may be and often is the case. However, the fact remains that planning and cost analysis seldom can effect economies in production in any degree comparable with those possible during basic design. It must be remembered that detail part drawings establish the number and complexity of fabricating steps required. Each feature and limiting specification beyond that necessary in the basic design increases total cost. By judicious forethought the designer can easily arrive at the most reasonable specifications.

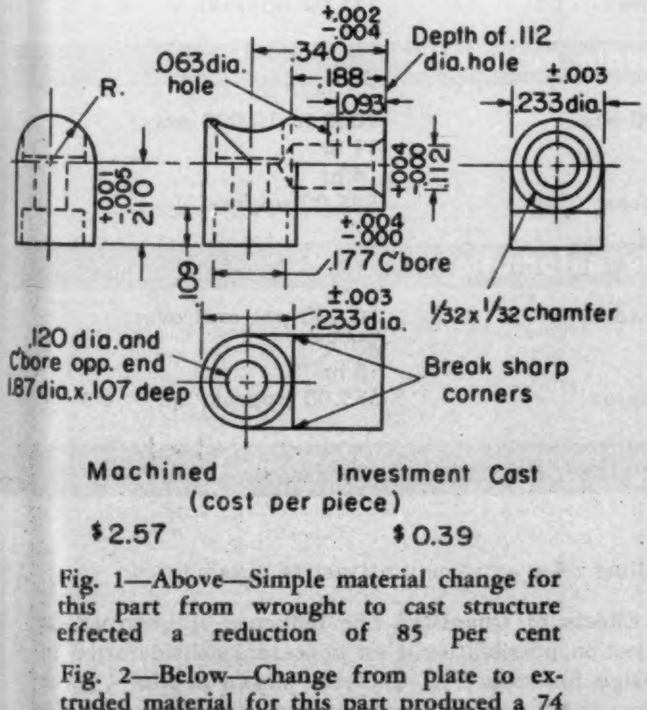
Processes: In many instances, any one or several of a group of processes may be satisfactory for production of a part without affecting the functional characteristics desired. For instance, the stainless steel elbow shown in *Fig.* 1 was designed for machining from bar stock. Specification change to permit cast rather than wrought material permitted a savings of $2.18 per piece.

The part illustrated in *Fig.* 2 was originally designed to be machined from plate. However, a specification change to permit use of an extruded shape

DCOSTS

How quantity, process and tolerance influence basic costs

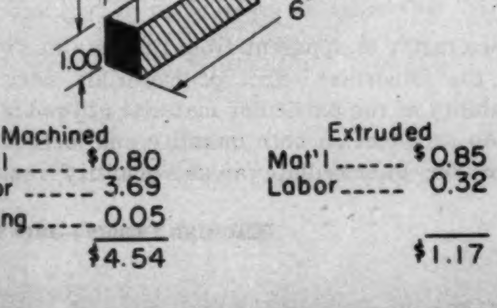
Engineering drawings of the designer set the limits on basic costs in manufacturing. General factors which affect costs can be controlled most readily by recognition of their character and scope in original design procedures



Machined	Investment Cast
(cost per piece)	
$2.57	$0.39

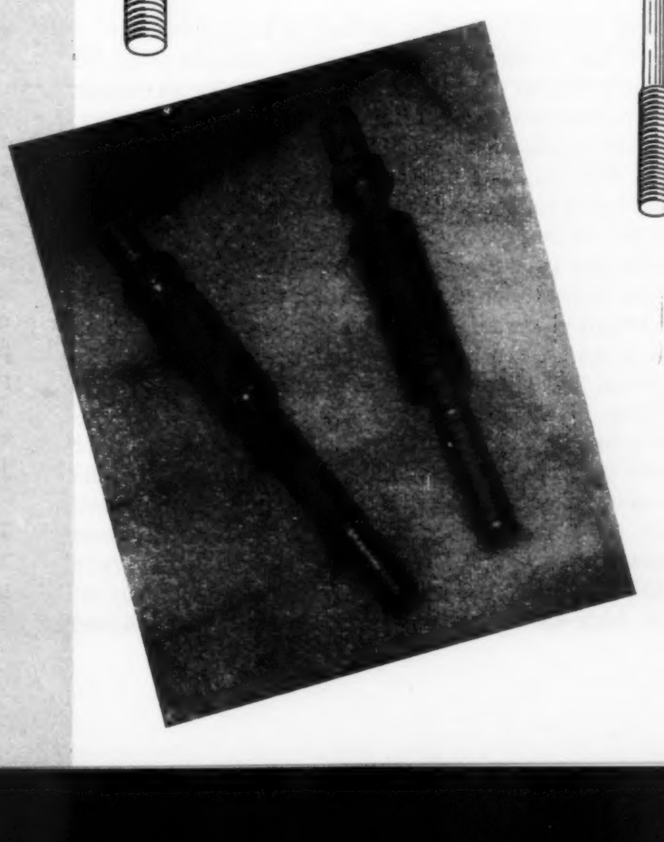
Fig. 1—Above—Simple material change for this part from wrought to cast structure effected a reduction of 85 per cent

Fig. 2—Below—Change from plate to extruded material for this part produced a 74 per cent saving in quantities of 2000 pieces



Machined		Extruded	
Mat'l	$0.80	Mat'l	$0.85
Labor	3.69	Labor	0.32
Tooling	0.05		
	$4.54		$1.17

Fig. 3 — Original steel screw machine produced, stand-off stud, left, cost $52.50 per thousand but redesigned unit with upset and roll threaded stud with a standard roll-formed spacer, right, cost only $10.50 per thousand



THREADED BUSHING

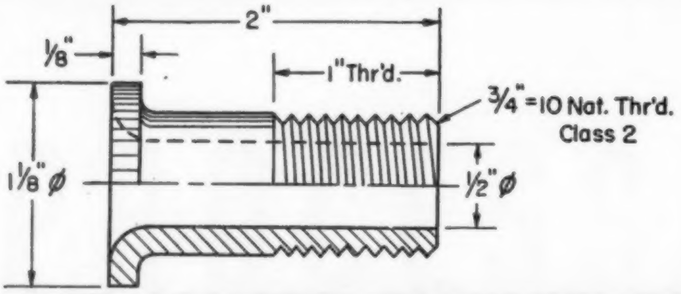


Fig. 4 — Threaded bushing with detailed costs for production by three methods and from three different materials

	Aluminum 17ST	Bar Steel B-1113	Bar Brass SAE 72
Materials	235 lb @ 0.41/lb = $96.50	656 lb @ 0.076/lb = $49.90	720 lb @ 0.3385/lb = $244.00
Turnings	170 lb @ 0.05/lb = $8.50		520 lb @ 0.175/lb = 91.00
Net material cost per M pcs	= $88.00	= $49.50	= $153.00
Method A	**Turret Lathe**		
Quantity	Under 1000 pcs	Under 250 pcs	Under 1000 pcs
Time per M	25 hr	61 hr	22 hr
Setup time	4 hr	4 hr	4 hr
Tooling Cost	$25.00 approx.	$25.00 approx.	$25.00 approx.
Method B	**Single-spindle automatic***		
Quantity	1000 to 2500 pcs	250 to 5000 pcs	1000 to 10,000 pcs
Time per M	21 hr	55 hr	11 hr
Setup time	4 hr	5 hr	4 hr
Tooling cost	$75.00 approx.	$45.00 approx.	$45.00 approx.
Method C	**Six-spindle automatic†**		
Quantity	2500 pcs and up	5000 pcs and over	10,000 pcs and over
Time per M	4.2 hr	7 hr	2.7 hr
Setup time	16 hr	16 hr	16 hr
Tooling cost	$150.00 approx.	$125.00 approx.	$75.00 approx.

* One operator can handle three to four machines. † One operator can handle two machines.

brought about a reduction in cost of $3.37 per piece.

With the foregoing parts, design changes required are insignificant but instances are equally numerous wherein considerable difference in design is necessary. One such is shown in *Fig.* 3. Here the functional end use is identical, but utterly different basic processing is employed to obtain the major decrease in cost with identical or improved strength.

With the part designed as originally shown in *Fig.* 3*a*, cost is established within rather firm limits. Quantity required usually indicates the process by which such a design can be most economically produced; in this case, 200,000 per year were produced on the screw machine. With large quantities, multiple-spindle machines can be used effectively but on quantities under 10,000 pieces a single-spindle machine might be the necessary choice for production. If quantity dropped under 1000 parts, a turret lathe might be the best means for turning. However, because quantity falls in the high-output bracket, processing by means of such high-speed methods as heading and thread rolling offer extremely attractive possibilities.

Effects of Quantity: The influence of quantity on selection possibilities is an important consideration in design for economy. The case shown in *Fig.* 4 gives some indication of how quantity would influence the method of machining a part. Because of tooling and setup costs as well as overhead, quantity is important in determining the most suitable method. Use of high-output machines such as multiple-spindle screw machines also involves the factor of efficiency. It often requires a day or two of production for such a machine to "settle down" and for an operator to get all the tools adjusted properly and become accustomed to particular machining characteristics of the job.

Another factor is apparent from the data in *Fig.* 4. This is the important effect of materials selection. Machinability of the particular material utilized is important in its effect on both quantity and method.

Except for intermediate areas, quantity require-

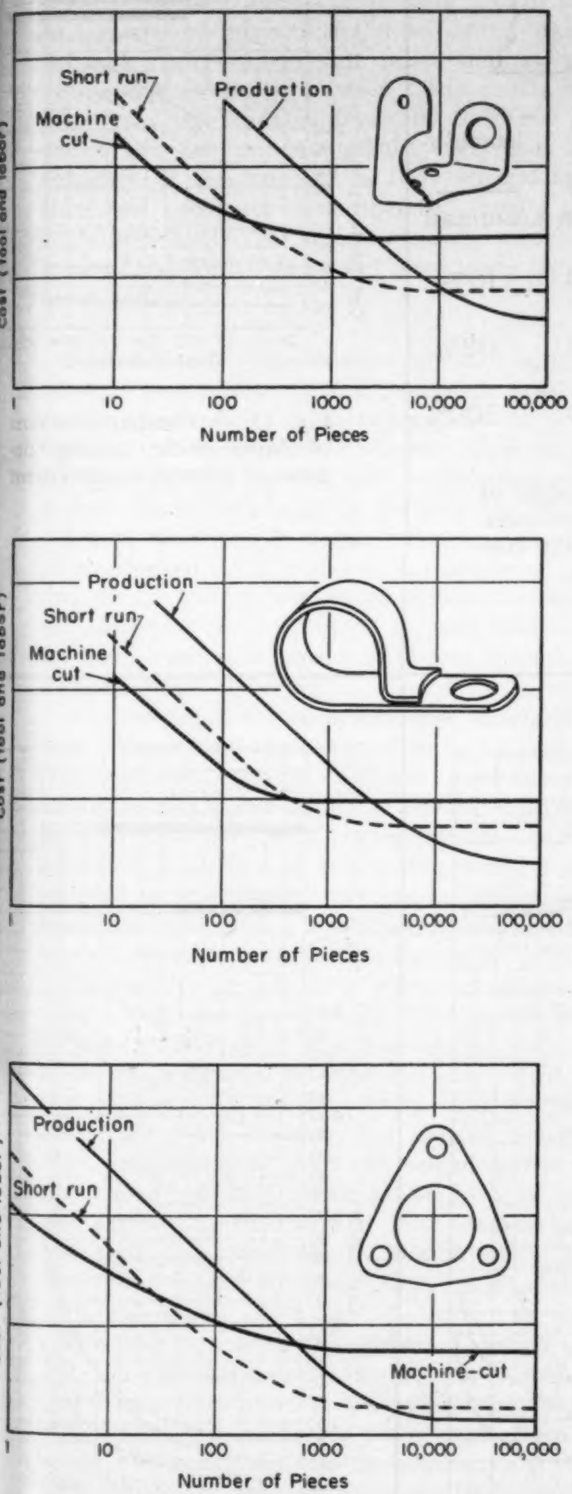


Fig. 5—Charts showing quantity-cost relationship for three parts designed for stamping produced by machine cutting, short-run and high-production methods

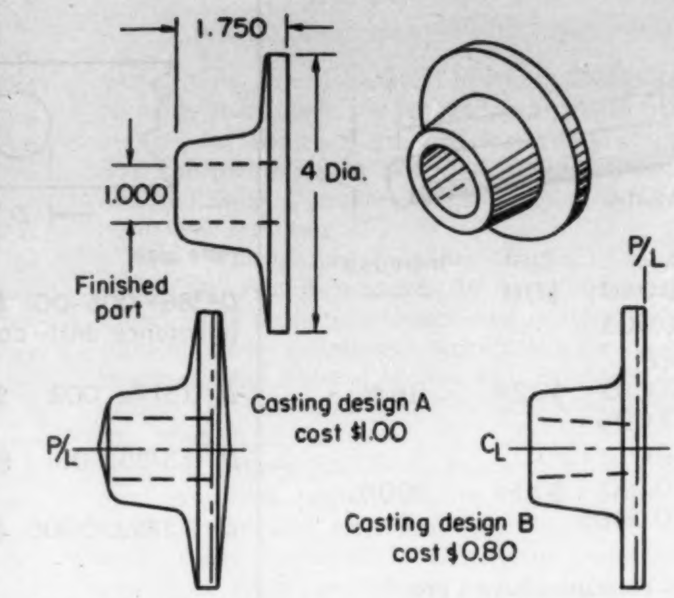


Fig. 6—Additional cost for magnesium casting *A* is due to the need for a core to produce the hole. Since the finished hole requires machining, the design as at *B* with draft in the hole offers a saving

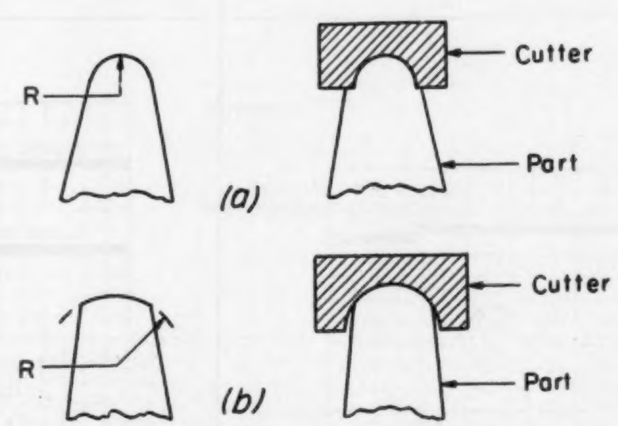
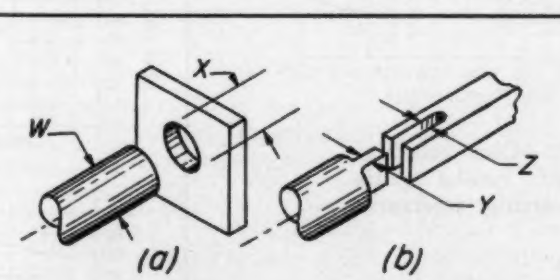
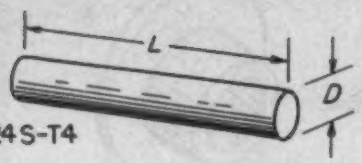


Fig. 7—To produce a design with a tangent radius, a concave cutter usually will create a nick on one or both sides of the piece as shown at *a*. The nontangent design shown at *b* offers considerable production economy



	Good	Poor
Nominal Dim. *W*	0.873	0.875
Nominal Dim. *X*	0.875	0.877
Nominal Dim. *Y*	0.239	0.250
Nominal Dim. *Z*	0.250	0.261

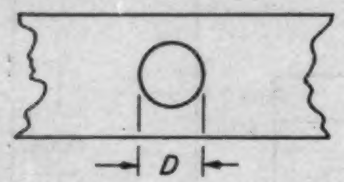
Fig. 8—Good and poor design of fitted parts such as these is indicated by the need for special cutters which incur added costs



Material: 24S-T4

		Cost	Increase
	Design A		
L =	3.00 ± 0.040	$0.11	——
D =	.50 ± 0.003		
	Design B		
L =	3.00 ± 0.010	$0.24	118%
D =	.495 ± 0.005		
	Design C		
L =	3.00 ± 0.010	$0.34	200%
D =	.4950 ± 0.0005		

Fig. 9—Tolerances have a profound effect on costs. Design *A* permits use of stock as drawn, *B* requires turning and *C* necessitates finish grinding



Hole Size	Cost	Increase
D=.386+.005-.001 (clearance drill—common tolerances)	$4.00	——
D=0.377±.002	$4.40	10%
D=0.3755±.001	$9.50	112%
D=0.375±.0005	$16.10	300%

Fig. 10—Cost relationship of holes with varying tolerances. Cost shown was for 120 consecutive holes

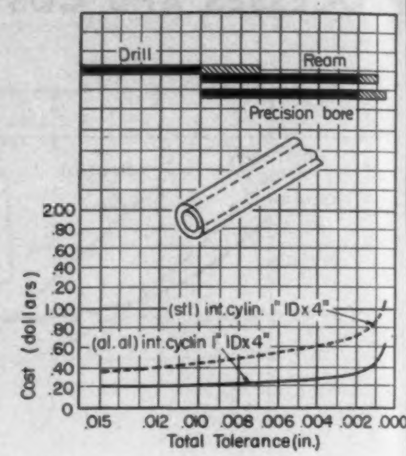


Fig. 11 — Comparative cost of holes with varying degrees of tolerance refinement

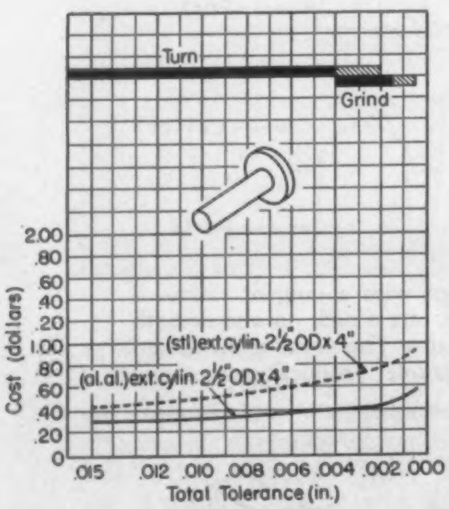


Fig. 12—Cost relationship of turned parts with varying tolerances

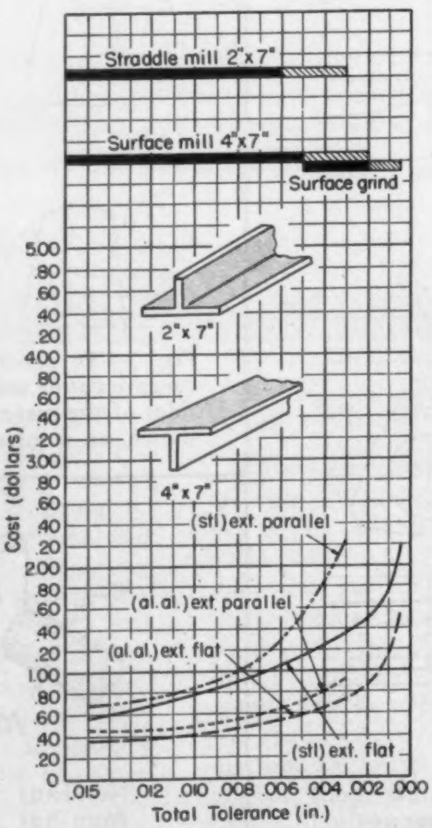


Fig. 13—Straddle and face milling costs plotted against tolerance requirements

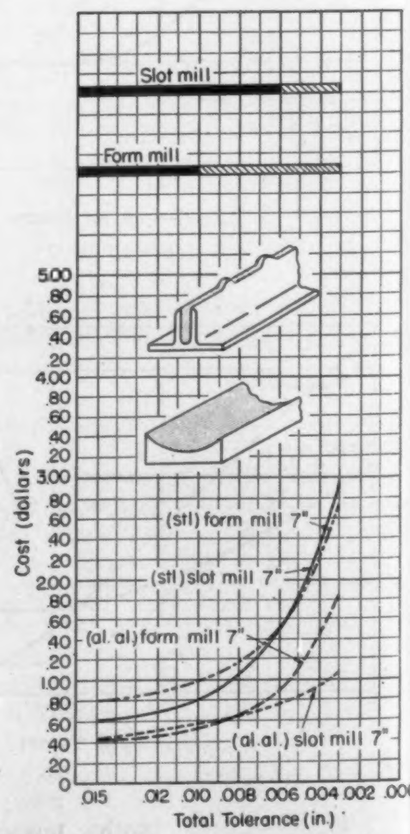


Fig. 14—Tolerance effect on cost of slotting and form milling cuts

ments are generally good indicators of suitable production methods. In these intermediate areas, all factors must be weighed carefully to assure clear-cut indication of where "breakeven" points occur. For instance, where stamping operations are concerned the range of possibilities runs from machine-cut, through short-run to full production stamping on high-speed automatic presses. The charts shown in *Fig.* 5 indicate an area from 100 to 10,000 pieces in which this occurs. As quantity increases from the category of machine-cut to fully automatic production, tool and setup costs increase rapidly but labor costs decrease. Tool costs are easily amortized over a large number of parts and labor costs per part become negligible.

Design Details: Specification of minor design details on drawings is also critical regarding possible processes and suitability of parts for an indicated process. Specification of certain details may render a part difficult to produce by the most economical method or require manufacture by a method less than ideal.

Small variations in design requirements often may be overlooked. For instance, the choice of a parting line for the part in *Fig.* 6—as dictated by whether or not allowance for draft could be permitted in the center hole—results in considerable cost difference owing to the method of coring.

Another small but recurrent case concerns machining. Where radii are designed to be tangent to adjacent surfaces, cutters used will most often fail to match perfectly and scrap or rework incurred is expensive, especially with critical stressed parts. This condition is illustrated at *a* in *Fig.* 7 and a preferred solution is shown at *b*, with the tangent or matching condition eliminated. Machining as shown at *Fig. 7a* costs three to four times that for one as detailed at *b*.

Dimensions as specified often necessitate special tools which incur added costs. The case shown in *Fig.* 8 shows two types of accurate, mating parts with dimensions designated as good and bad. While little if any difference in manufacturing time would be evidenced in either instance, the special tools required must be considered. With the hole at *a, Fig.* 8, a special reamer, costing about $25.00, was necessary to hold the 0.877-inch diameter. For the slotting operation at *b*, to produce the 0.261-inch width, a special milling cutter costing about $50.00 was needed.

Tolerances: Among the effects of design specifications on costs, those of tolerances are perhaps most significant. Tolerances in design influence the producibility of the end product in many ways, from necessitating additional steps in processing to rendering a part completely impractical to produce economically. In range, tolerances may cover dimensional variation, surface roughness range, allowable variation in such properties as heat treatment, endurance time of coatings against corrosion, etc.

DIMENSIONAL LIMITS: A simple example of how tolerances can increase cost is given in *Fig.* 9. Tolerances on dimensions specified at values smaller than functional requirements actually necessitate, increase costs through additional machine time, checking and gaging time, rejections, etc. In *Fig.* 9, Design *A* can be made from bar stock by sawing to length but tolerances for Design *B* are too close for sawing. To hold the tolerances on Design *B*, turning is required and for Design *C* grinding is necessary at an increase of 200 per cent over *A*.

In the same manner holes are affected by specifications. The cost data in *Fig.* 10 were found from an actual study made to determine the relative effects of tightening hole tolerances. Some 120 holes were used in determining the cost factors and these data do not include the additional costs of extra tooling, gages, etc., necessary.

Again, drilling to close tolerances is more costly in hard materials, materials with inclusions or hard spots, and in thin materials under ⅛-inch in thickness. Holes over 1-inch in diameter or four times diameter in depth are difficult to hold to close tolerances and costs increase rapidly if extreme precision is demanded. *Fig.* 11 shows the comparative costs of holes with varying degrees of refinement. The solid

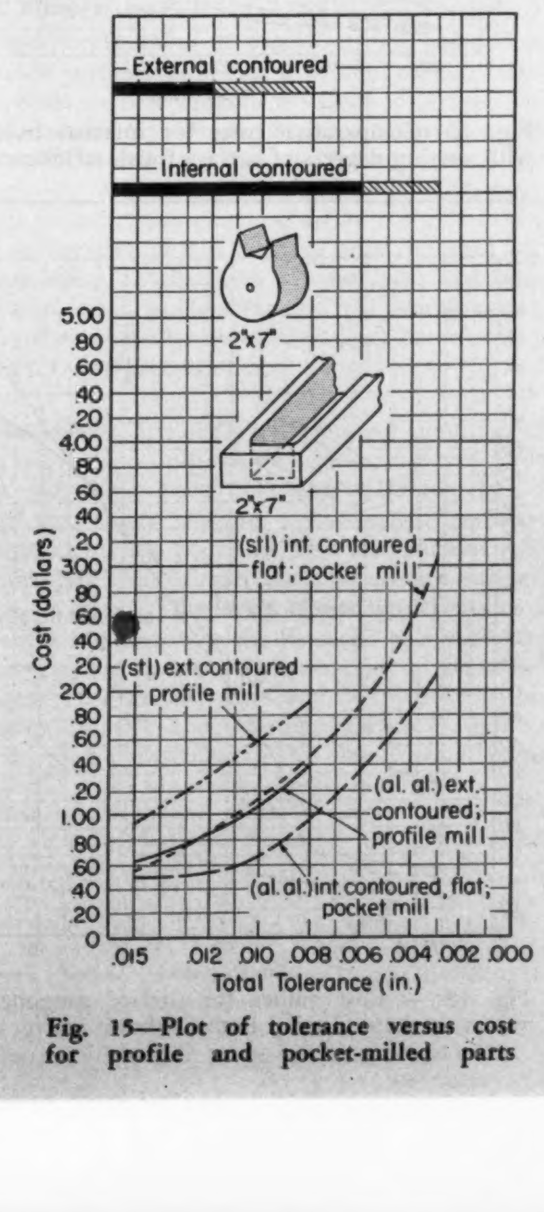


Fig. 15—Plot of tolerance versus cost for profile and pocket-milled parts

bar indicates the most desirable range of values.

Some idea of costs relative to turning can be gained from the curves of *Fig.* 12. Long parts of relatively small diameter subject to deflection in turning are generally difficult to produce. Recesses requiring long or unsupported tools are also difficult to machine to close tolerances. Tapers, contours, large radii, and related areas too wide for single form tools require special machines or tools to achieve fine limits.

Straddle and face milling costs are illustrated in *Fig.* 13. Straddle milling with heavy cuts or hard materials may cause cutter deflection on entry and exit of cuts. Close tolerances may require additional cuts to remove taper. With face milling some heat warpage can result with the necessity of additional slow light "skin" cuts to achieve accuracy desired. With slotting and forming cuts, the basic form tolerance is ground into the tool. Close tolerances, however, can result in short cutter life between grinds with greatly increased maintenance and tool costs. *Fig.* 14 gives a comparative cost evaluation. Excessively close tolerances on slotting can necessitate a minimum of three passes—one rough and two finish—to hold size. Profiling and pocket milling costs are given in *Fig.* 15.

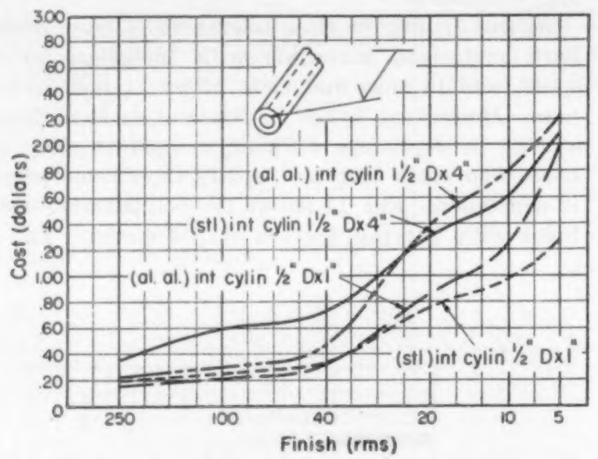


Fig. 16—Comparative costs for internal holes with varying degrees of surface finish refinement

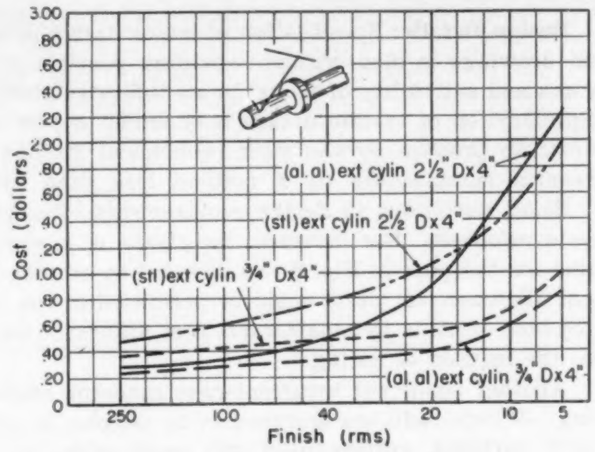


Fig. 17—Plot of surface refinement versus cost for some turned and ground surfaces

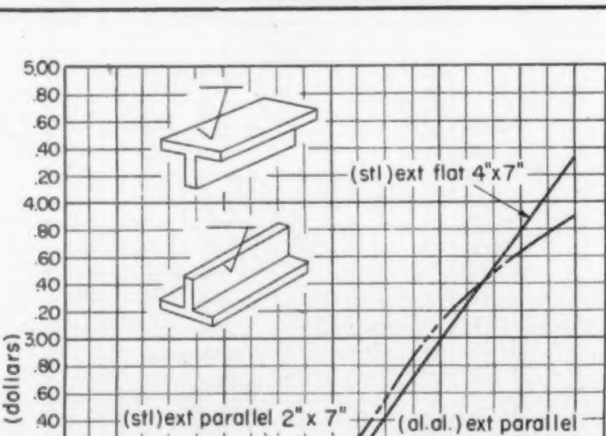


Fig. 18 — Cost values for surface roughness values on shapes similar to those shown in *Fig.* 13

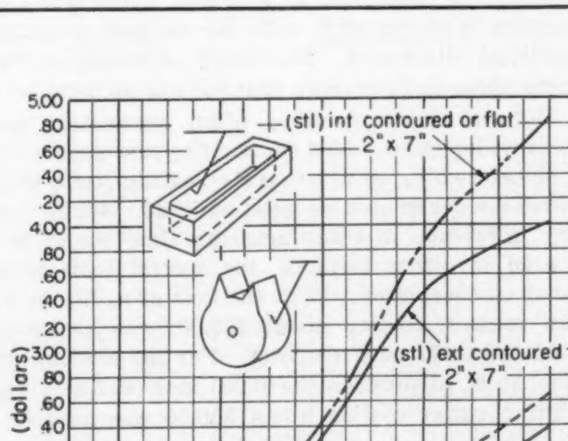


Fig. 19—Surface finish versus cost of production for profiled and pocket-milled parts

Here, tolerances are greatly limited by tendency of the cutter to spring from the work. Numerous passes and light cuts are necessary for close limits and on enclosed surfaces smooth blending of surfaces is difficult when the direction of cut changes.

Surface Roughness Limits: Finish demands impose severe cost problems. As surfaces are refined, tool costs rise. Cutting tools used to produce a 250 rms surface roughness will last twice as long in production, before sharpening, as those required for a 100 rms finish. Those for 100 rms roughness will last three times as long as tools needed to hold a 40 rms finish.

Below a 40 rms surface it is generally necessary to move work from general machines to precision equipment incurring greater costs. In addition, finer finishes add to total costs owing to increased inspection time, special handling and susceptibility to damage.

A general picture of comparative costs for internal cylindrical holes is given in *Fig.* 16. To hold costs to a minimum it is suggested that finish on internal cylindrical surfaces to be held to the following limits:

Drill	To 250 rms minimum
Ream	To 100 rms minimum
Bore	To 40 rms minimum

The sharp upward break at 40 rms in *Fig.* 16 reflects the addition of grinding to attain finishes below this value.

Similar specification of widest possible limits on finish of external cylindrical surfaces, as well as on milled and profiled surfaces, is desirable. In *Fig.* 17 are plotted some cost figures on examples of turned and ground surfaces. *Fig.* 18 shows cost values for surface finish on shapes such as shown in *Fig.* 13. Here again, lowest cost is obtained by use of roughest suitable surface condition. Examples plotted in *Fig.* 19, similar to those of *Fig.* 15, cover end-milled and profiled surfaces as obtained without special attention.

Specific surface roughness values to be used on precision machinery components can often pose a problem. Without actual test or trial run, the allowable roughness may be difficult to establish. However, to assist in determining or evaluating suitable roughness values the data in *Fig.* 20 have been compiled. These values are typical of the roughnesses used on some precision machine elements and will permit evaluation of finishes normally obtainable from production processes for any particular application.

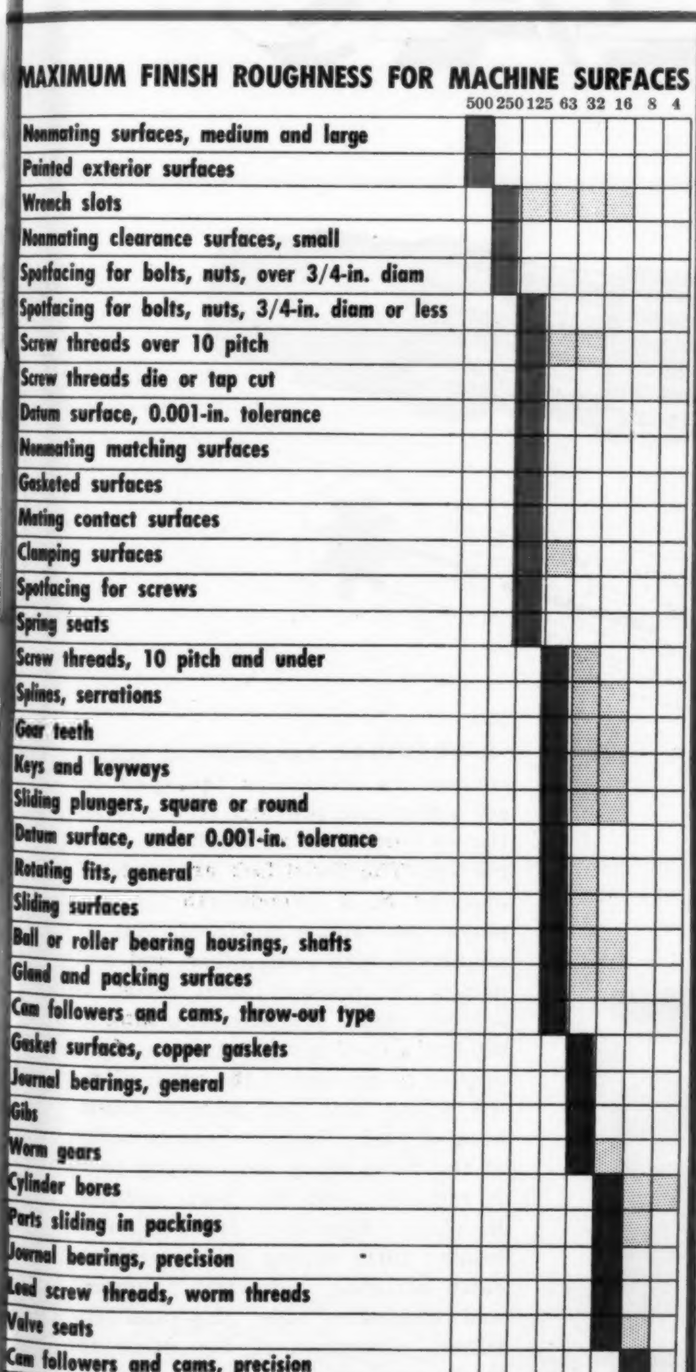


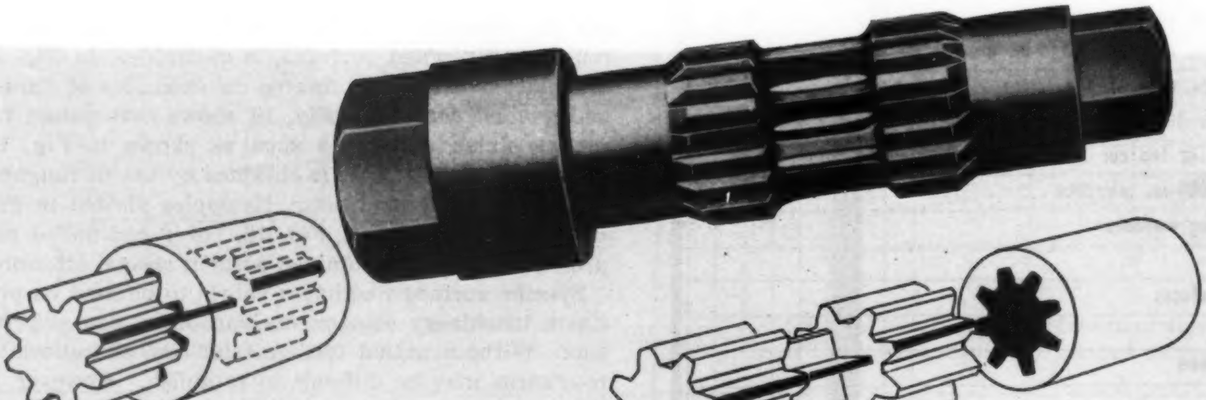
Fig. 20—Some typical surface roughness values specified for precision machine part surfaces

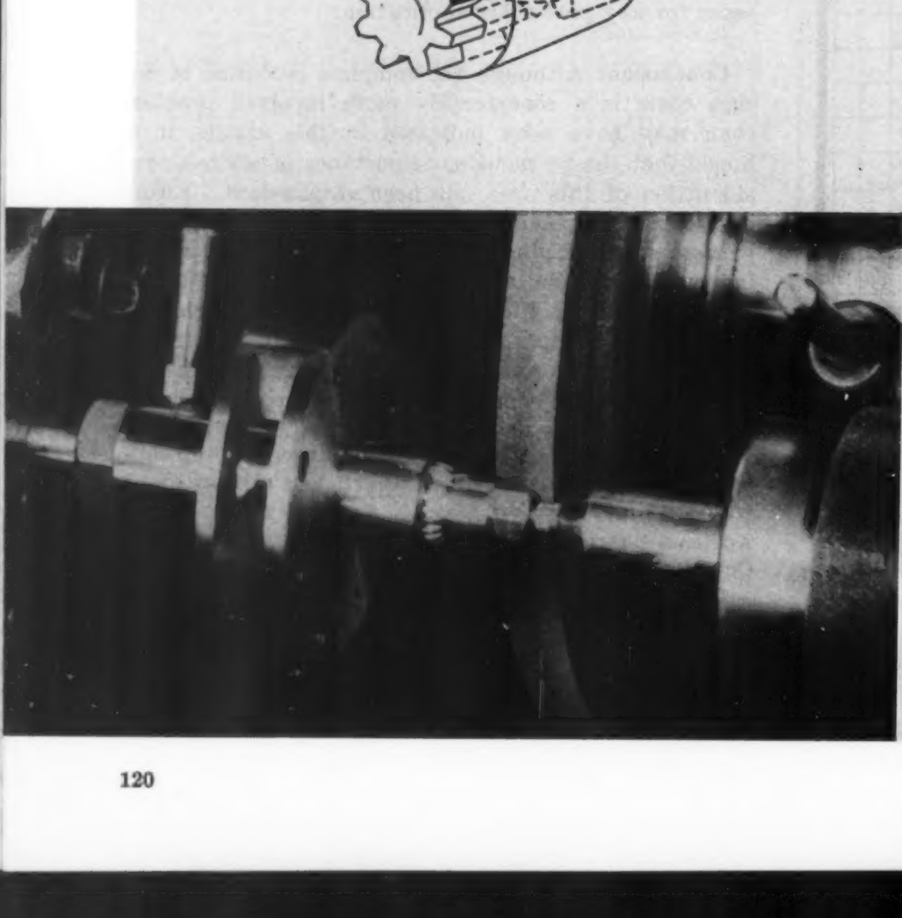
Conclusion: Although the complete evolution of design costs is a considerably more involved problem than may have been indicated in this article, it is hoped that the tremendous importance of serious consideration of this area has been emphasized. Effort and capital expenditure in streamlining production processes, simplifying steps in manufacture and speeding materials handling can be saved completely or reduced greatly by the elimination of all possible unnecessary operations. The key to low costs is in the designer's hand. Adequate information as well as incentive and programming by engineering management will produce unprecedented results.

Collaboration of the following organizations in supplying substantiating data for this article is acknowledged with much appreciation:

Austenal Laboratories Inc. (*Fig.* 1) .... Chicago, Ill.
Consolidated-Vultee Aircraft Corp. (*Figs.* 11, 12, 13, 14, 15, 16, 17, 18, and 19) .... San Diego, Calif.
General Electric Co. (*Fig.* 3) .... Schenectady, N. Y.
Laminated Shim Co., Stampings Div. (*Fig.* 5) .... Glenbrook, Conn.
Northrop Aircraft Inc. (*Figs.* 2, 6, 7, 8, 9, and 10) .... Hawthorne, Calif.

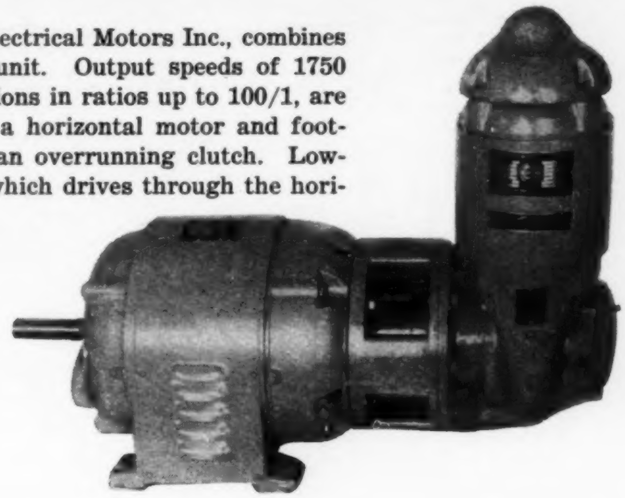
Scanning the field for IDEAS



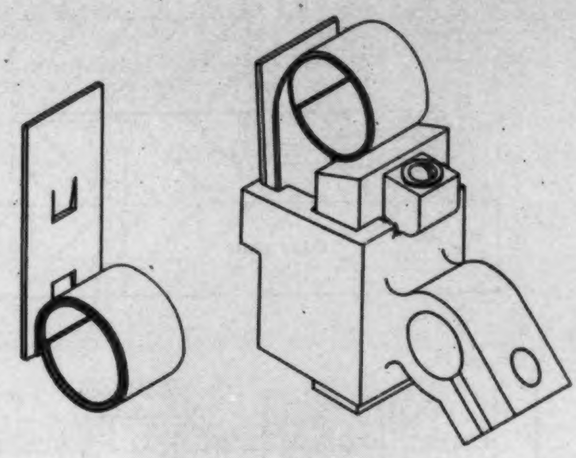


TORSIONAL LOCKING of splined connections eliminates the need for set screws and provides secure retention of parts with automatic backlash takeup. The Twist-Lok arbor, developed by N. A. Woodworth Co., employs two spline surfaces, out of alignment with each other and connected by resilient steel reeds, to achieve rapid, precision chucking of internal spined parts. A twisting force applied to one end of the arbor aligns the spline teeth and permits assembling of parts. Torsional spring action of the reeds when the twisting force is removed locks the parts securely in place. Equalized distribution of locking force among spline teeth assures accurate, concentric alignment and eliminates side play and end thrust. Operation of the arbor for loading and unloading may be manual or powered.

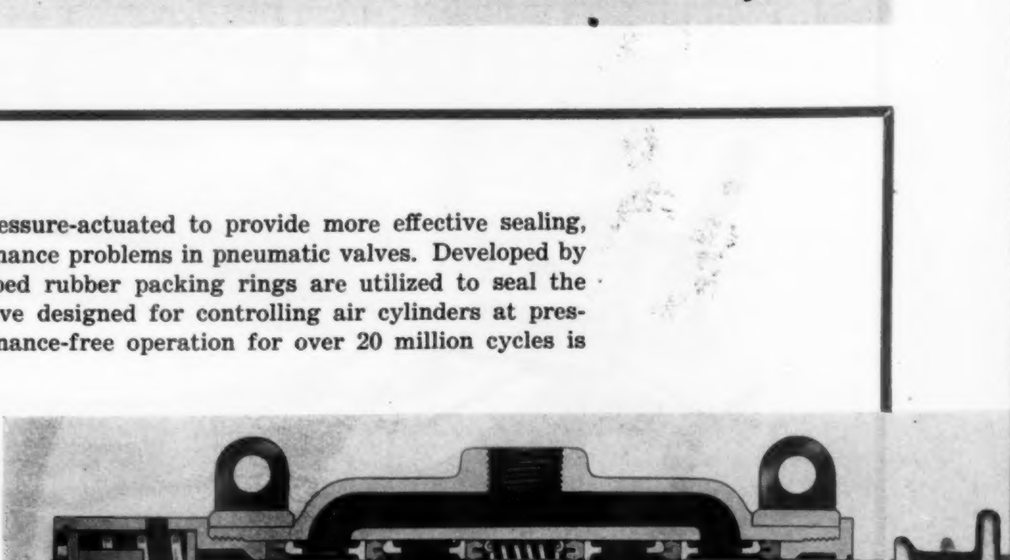
DUAL POWER DRIVE developed by U. S. Electrical Motors Inc., combines two motor performance in a single mounted unit. Output speeds of 1750 and 100 rpm, as well as other special combinations in ratios up to 100/1, are provided by a packaged design which utilizes a horizontal motor and footless right-angle gearmotor connected through an overrunning clutch. Low-speed operation is provided by the gearmotor which drives through the horizontal motor with the overrunning clutch engaged. For high speeds, the horizontal motor is energized and overruns the gearmotor. Designed primarily to meet two-speed cycle requirements in applications such as dry cleaning machinery, the unit is silent in operation and offers simplified installation through one mounting base.

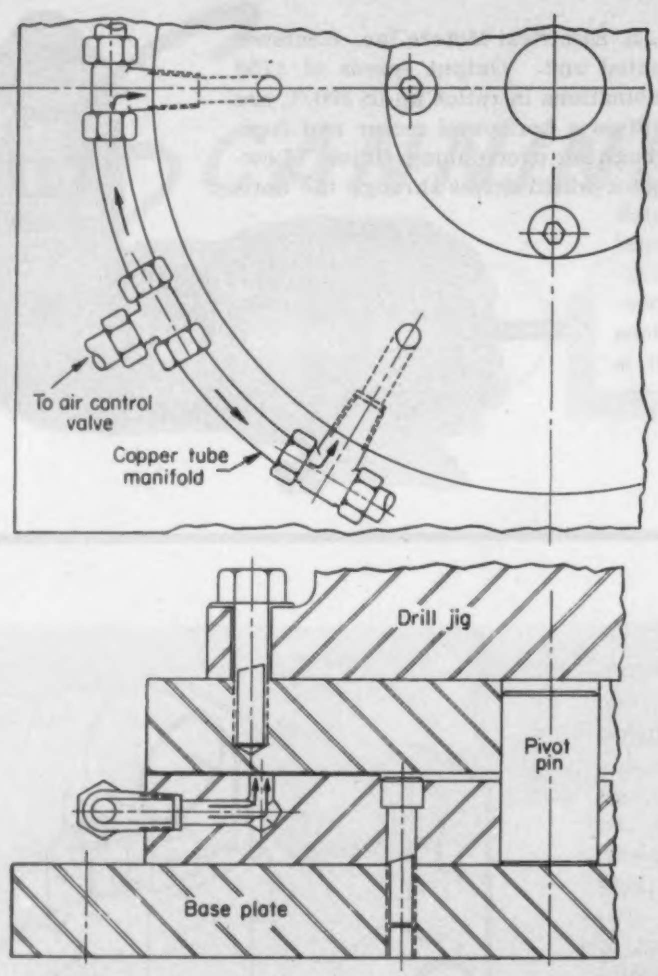


NONVARYING BRUSH PRESSURE LOADING for electric motors improves commutation and obviates resetting to compensate for brush wear. In the assembly developed by Hunter Spring Co., an extension neg'ator spring fastened to a backup plate, which snaps in and out of position without tools, acts as the brush pressure element and provides a constant force on the brush regardless of the extent of wear. Arcing between brush and commutator is minimized and the smaller number of parts required as compared to conventional designs reduces crowding between mounted sets.

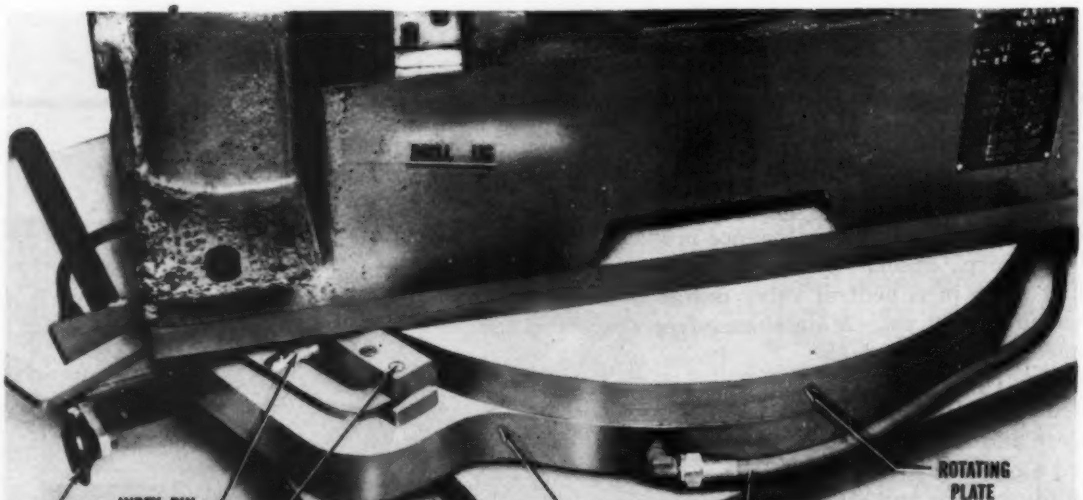


FLOATING PACKINGS, pressure-actuated to provide more effective sealing, simplify assembly and maintenance problems in pneumatic valves. Developed by Valvair Corp., special U-shaped rubber packing rings are utilized to seal the sliding stem in a neutral valve designed for controlling air cylinders at pressures up to 300 psi. Maintenance-free operation for over 20 million cycles is claimed. Position of the packing rings is maintained by a series of brass spacers and a compression spring, eliminating the need for grooves and other surface discontinuities usually employed. Design of the spacers permits free air flow through the valve during operation.

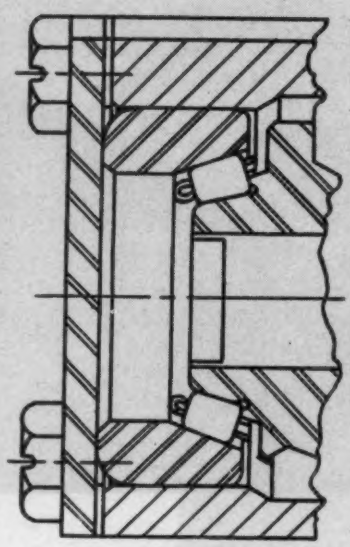




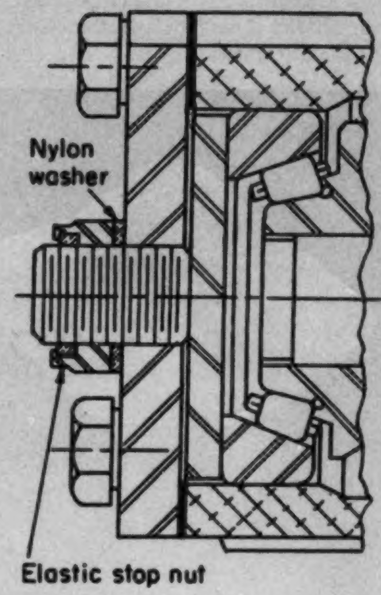
MINIMUM FRICTION in the rotation of heavy bed plates or machine tables for positioning or indexing operations is achieved with a unique hydrostatic bearing employing compressed air. In a design developed by Solar Aircraft Co. for a drill jig index table, air at 90 psi is supplied through a tube manifold and manual control valve arrangement to "float" the rotating plate approximately 0.005-inch above its fixed base, facilitating manual indexing and reducing wear of the contact surfaces. Side air leakage, which reduces the effective lifting pressure, is counteracted by using as-machined contact surfaces — the slight surface irregularities act to distribute and retain the air pressure, providing a pressure differential adequate for lifting action. Chip protection, a necessity for air bearings, is provided by felt wipers employed to seal the joint formed at the plate and its base. Cost of the air bearing is less than conventional thrust bearings, particularly where large or heavy jigs are being used.



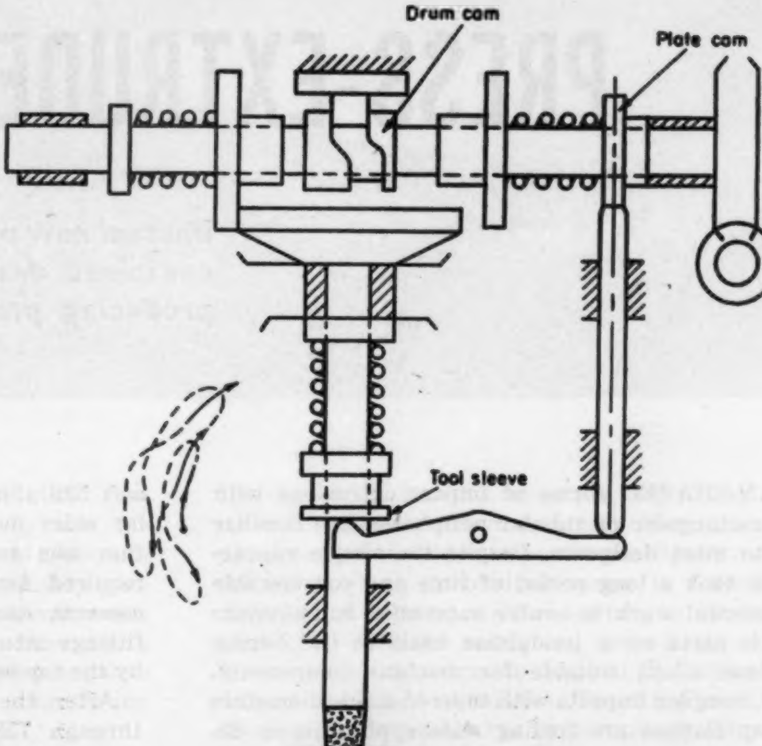
IDEAS



SETSCREW BEARING PRELOADING facilitates worm adjustment and simplifies assembly of the Hydraguide power steering unit. Developed by Gemmer Mfg. Co., the design employs a socket setscrew, Elastic Stop Nut and circular pressure plate, replacing a conventional assembly using shims. In the new design, accurate preloading of the tapered roller bearings is accomplished by adjusting the setscrew through a threaded hole in the bolted endplate, obviating the necessity for disassembly previously required. Retention of adjustment is assured by the locking action of the nut which also serves to seal the thread against loss of oil. Although actually increasing the number of parts required, the new design has resulted in appreciable ultimate gains through improved production and service.



IRREGULAR MOTION for lapping-in valve seats is produced mechanically in a machine designed by W. Beer, Wiesbaden, Germany. A unique cam and linkage arrangement transmits the rotary motion of a horizontal splined shaft to a lapping tool, sleeve mounted on a vertical shaft, to duplicate hand operations usually employed. Reciprocated by a drum cam, the driven horizontal shaft carries two spring-loaded disks which alternately engage a disk on the vertical shaft, oscillating the shaft about a vertical axis. Simultaneously, a plate cam on the horizontal shaft actuates a pushrod and lever to reciprocate the tool sleeve in a vertical plane. Result of the combined motion is a swinging tool movement of varying arc coupled with a small lifting action.



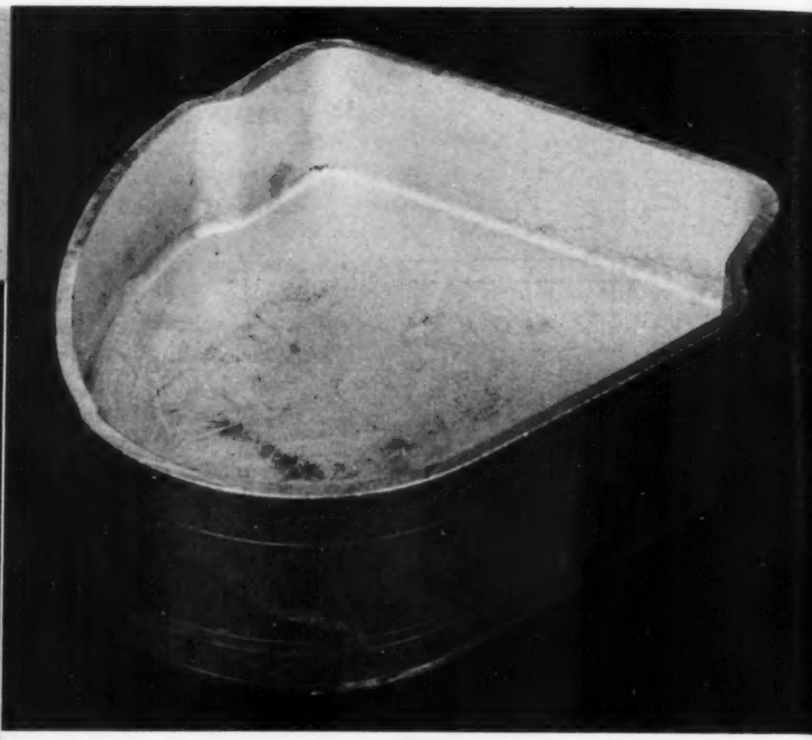


Fig. 1 — Asymmetrical impact extrusion for producing two aircraft shear tie fittings

By L. M. Christensen
Forging Design Engineer
Producibility Engineering Section
Northrop Aircraft Corp.
Hawthorne, Calif.

PRESS-EXTRUDED FORGINGS

Unusual new possibilities are offered the cost conscious designer by a new process for producing precision forgings without draft

CAN-SHAPED forms of impact extrusions with rectangular or tubular peripheries are familiar to most designers. Despite the simple appearance, it took a long period of time and considerable experimental work to evolve successful manufacture of such parts on a production basis in the harder aluminum alloys suitable for machine components. Today, complex impacts with tapered inside diameters and cap flanges are finding wide application in design.

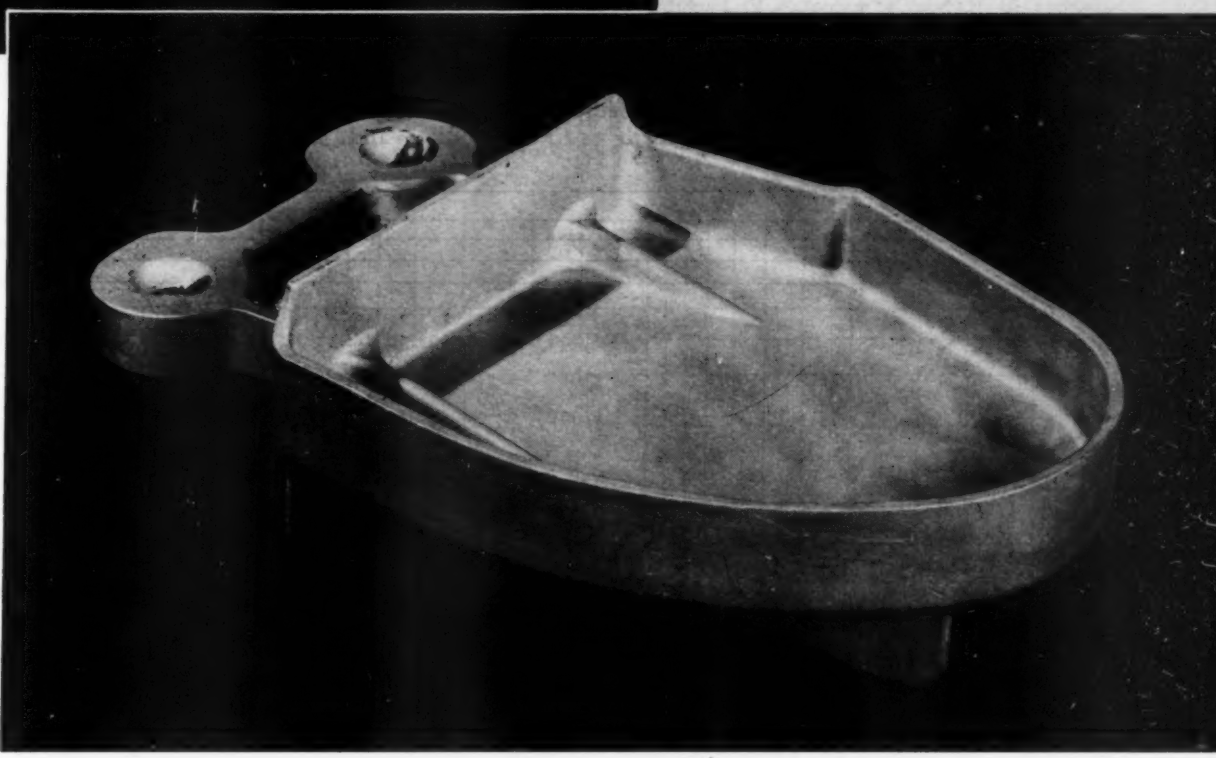
The early conception of impact extrusions was that the parts had to be of a symmetrical nature. Northrop Aircraft's first venture into asymmetrical shapes was the part shown in *Fig.* 1, of relatively soft 52S aluminum alloy, which not only had irregular sides but had varying wall thickness. Application was such that obtuse and acute angles were required for typical shear tie fittings, and by the common concept of making a multiple number of fittings into a cup, the peculiar geometry indicated by the top section was evolved.

After the war, use of harder alloys—from 14S through 75S—for the simpler shapes was accomplished with considerable research and difficulty. Lubrication of dies was a big problem and directly and indirectly attributed to various troubles such as having the part squeeze up like an accordion when an attempt was made to extract it from the

Fig. 2—Left—Heavy-wall aileron nose rib produced by impact extrusion

Fig. 3—Left — Complex aileron nose rib as produced by conventional forging methods from 14ST aluminum alloy

Fig. 4—Below—Aileron nose rib design produced as a press extruded forging. Side walls have no draft



punch. With the development of hard alloy can-shaped parts, impact extrusion usage by the air-frame designer increased greatly owing to close tolerances available complemented by high-quality surfaces and elimination of draft angles. Draft angle on conventional forgings is more often than not a nuisance and, consequently, straight-sided impacts are valuable in both weight reduction and tooling simplification.

Early Developments: Not long after World War II, one of the leading forging companies took a forward step along the lines of improving and extending impact-extrusion techniques by the design and fabrication of the part shown in *Fig.* 2, an aileron nose rib formerly drawn to shape. Due to loads imposed, metal thickness was necessarily increased to the point where it could no longer be formed from sheet metal and, to avoid machining a conventional forging on both the inside and the outside surfaces, it was decided to fabricate this and five other similar designs as impact extrusions. Resulting parts suited the application admirably by completely eliminating all Hydrotel machining operations and reducing the conventional machining operations to a minimum. Accuracy of contours was brought to plus or minus 0.015-inch and, in addition, the quality of the mating surfaces was equivalent to 100 rms with no apprecia-

By L. M. Christensen

Forging Design Engineer
Producibility Engineering Section
Northrop Aircraft Corp.
Hawthorne, Calif.

PRESS-EXTRUDED FORGINGS

Unusual new possibilities are offered the cost conscious designer by a new process for producing precision forgings without draft

CAN-SHAPED forms of impact extrusions with rectangular or tubular peripheries are familiar to most designers. Despite the simple appearance, it took a long period of time and considerable experimental work to evolve successful manufacture of such parts on a production basis in the harder aluminum alloys suitable for machine components. Today, complex impacts with tapered inside diameters and cap flanges are finding wide application in design.

The early conception of impact extrusions was that the parts had to be of a symmetrical nature. Northrop Aircraft's first venture into asymmetrical shapes was the part shown in *Fig.* 1, of relatively soft 52S aluminum alloy, which not only had irregular sides but had varying wall thickness. Application was such that obtuse and acute angles were required for typical shear tie fittings, and by the common concept of making a multiple number of fittings into a cup, the peculiar geometry indicated by the top section was evolved.

After the war, use of harder alloys—from 14S through 75S—for the simpler shapes was accomplished with considerable research and difficulty. Lubrication of dies was a big problem and directly and indirectly attributed to various troubles such as having the part squeeze up like an accordion when an attempt was made to extract it from the

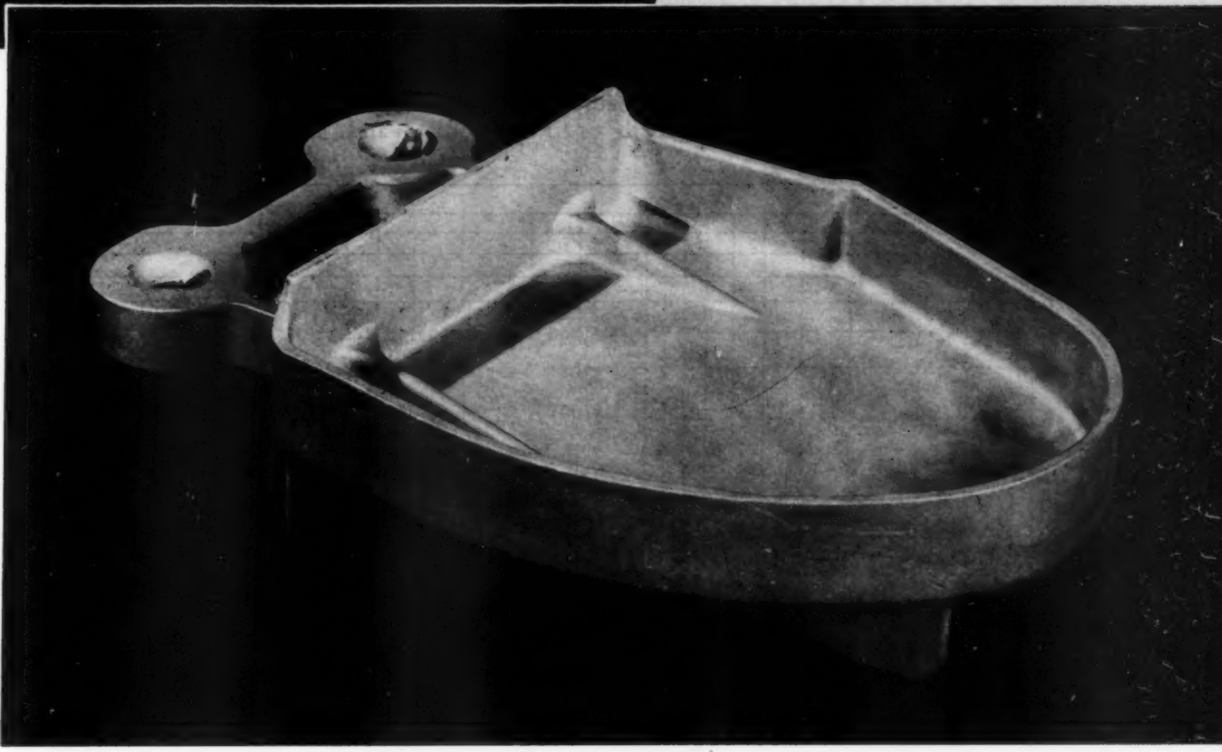
ble die drag marks or scoring.

One part proved particularly troublesome. Forging with a conventional channel type design having a 5 degree draft angle was attempted with a subsequent ironing operation to eliminate the objectionable draft angle. This experiment, however, proved unsuccessful, and this one of the six designs had to be abandoned in favor of a conventional forging. Despite successful use of the other five designs in production, fabrication troubles occurred in manufacture and though some progress was made it was not of a sensational nature.

Extrusion Forging: In pursuing this problem further, contact was made with Prex Forgings Corp. who had, by a new method of extrusion forging, made some parts which looked very promising. This method of manufacture is predicated on forming metal in a hydraulic press, with apparently no draft angle, in a manner somewhat similar to extrusion and is termed by Prex as "press extruded forging."

Following consultation, a detailed cost study of the latest aileron nose rib was made to show the comparative figures for the part as machined from a billet or from a conventional forging, and as a press-extruded forging. In *Fig.* 3 is shown the conventional forging from which the part was being processed while *Fig.* 4 shows the product as Prex proposed to make it. The thin flanges, without draft angle, and the close tolerances of the press-extruded forging eliminated virtually all machining. The chart of *Fig.* 5 shows comparative costs of the three methods. It is interesting to note that the press extruded form costs less than either the machined billet or regular forging at considerably less than 100 parts. On the basis of this cost study, an order was placed with Prex for development and fabrication of production quantities by the press-extruded method.

After completion of the dies a certain amount of further work was necessary since this part was considerably more difficult than any tried before. However, sample parts were completed on schedule, and after some additional refinements, first production shipment was sent in. The job has now progressed



punch. With the development of hard alloy can-shaped parts, impact extrusion usage by the airframe designer increased greatly owing to close tolerances available complemented by high-quality surfaces and elimination of draft angles. Draft angle on conventional forgings is more often than not a nuisance and, consequently, straight-sided impacts are valuable in both weight reduction and tooling simplification.

Early Developments: Not long after World War II, one of the leading forging companies took a forward step along the lines of improving and extending impact-extrusion techniques by the design and fabrication of the part shown in *Fig.* 2, an aileron nose rib formerly drawn to shape. Due to loads imposed, metal thickness was necessarily increased to the point where it could no longer be formed from sheet metal and, to avoid machining a conventional forging on both the inside and the outside surfaces, it was decided to fabricate this and five other similar designs as impact extrusions. Resulting parts suited the application admirably by completely eliminating all Hydrotel machining operations and reducing the conventional machining operations to a minimum. Accuracy of contours was brought to plus or minus 0.015-inch and, in addition, the quality of the mating surfaces was equivalent to 100 rms with no apprecia-

Table 1—Physical Properties of Press Extruded Forgings

Forging No.	Specimen No.	Yield Strength (psi, 0.2% offset)	Tensile Strength (psi)	Elong. (% in 2")	Hardness (Rock. B)
1	1	61,670	65,520	3.0	81–81
1	2	59,590	65,670	7.5*	79–80
1	3	62,500	69,240	10.0	81–81
1	4	60,810	67,340	7.0	80–80
1	5	61,780	68,530	7.0	80–81
1	6	57,970	66,370	7.0	79–81
1	7	59,460	66,370	7.0*	80–81
1	8	57,890	66,470	9.0	77–78
2	1	61,660	67,770	5.0	80–81
2	2	60,450	68,990	8.0*	80–82
2	3	61,710	69,030	8.0	80–82
2	4	61,860	68,730	7.0	81–81
2	5	63,580	69,940	8.0	81–82
2	6	61,110	68,160	7.0	81–81
2	7	60,030	67,420	7.0*	80–81
2	8	60,590	67,760	9.0	81–81

* Denotes 1-inch gage length.

Note: AMS-4135G minimum requirements per paragraphs 5.1.1.1 and 5.1.1.2 are as follows:

Tensile Strength Yield Strength Elong.

to the point where several hundred acceptable parts have been made.

From the first production shipment, two parts were chosen at random and subjected to a detailed physical and metallographic evaluation. Physical test specimens and metallographic specimens were cut from the parts as shown by *Fig.* 6. Particular interest was placed in obtaining specimens from areas which were known to be troublesome during the development of the forging techniques. From each part, a total of eight physical test specimens and four metallographic samples were tested with the results as shown in TABLE 1. These results were considered to be more than satisfactory since all portions of the part exceeded minimum specification properties as called for in the 14ST Forging Specification No. AMS4135.

Grain flow studies indicated excellent metal flow control with a resulting grain direction that was oriented in a most efficient manner. Upon viewing these grain flow studies, the inefficiency of machining away good grain flow characteristics is readily

ble die drag marks or scoring.

One part proved particularly troublesome. Forging with a conventional channel type design having a 5 degree draft angle was attempted with a subsequent ironing operation to eliminate the objectionable draft angle. This experiment, however, proved unsuccessful, and this one of the six designs had to be abandoned in favor of a conventional forging. Despite successful use of the other five designs in production, fabrication troubles occurred in manufacture and though some progress was made it was not of a sensational nature.

Extrusion Forging: In pursuing this problem further, contact was made with Prex Forgings Corp. who had, by a new method of extrusion forging, made some parts which looked very promising. This method of manufacture is predicated on forming metal in a hydraulic press, with apparently no draft angle, in a manner somewhat similar to extrusion and is termed by Prex as "press extruded forging."

Following consultation, a detailed cost study of the latest aileron nose rib was made to show the comparative figures for the part as machined from a billet or from a conventional forging, and as a press-extruded forging. In *Fig.* 3 is shown the conventional forging from which the part was being processed while *Fig.* 4 shows the product as Prex proposed to make it. The thin flanges, without draft angle, and the close tolerances of the press-extruded forging eliminated virtually all machining. The chart of *Fig.* 5 shows comparative costs of the three methods. It is interesting to note that the press extruded form costs less than either the machined billet or regular forging at considerably less than 100 parts. On the basis of this cost study, an order was placed with Prex for development and fabrication of production quantities by the press-extruded method.

After completion of the dies a certain amount of further work was necessary since this part was considerably more difficult than any tried before. However, sample parts were completed on schedule, and after some additional refinements, first production shipment was sent in. The job has now progressed

Fig. 5 — Chart showing plot of comparative cumulative costs for the aileron nose rib as produced by three methods. Crossover points are: methods 2 and 3, after 66th part; methods 1 and 2, after 69th part; methods 1 and 3, after 68th part

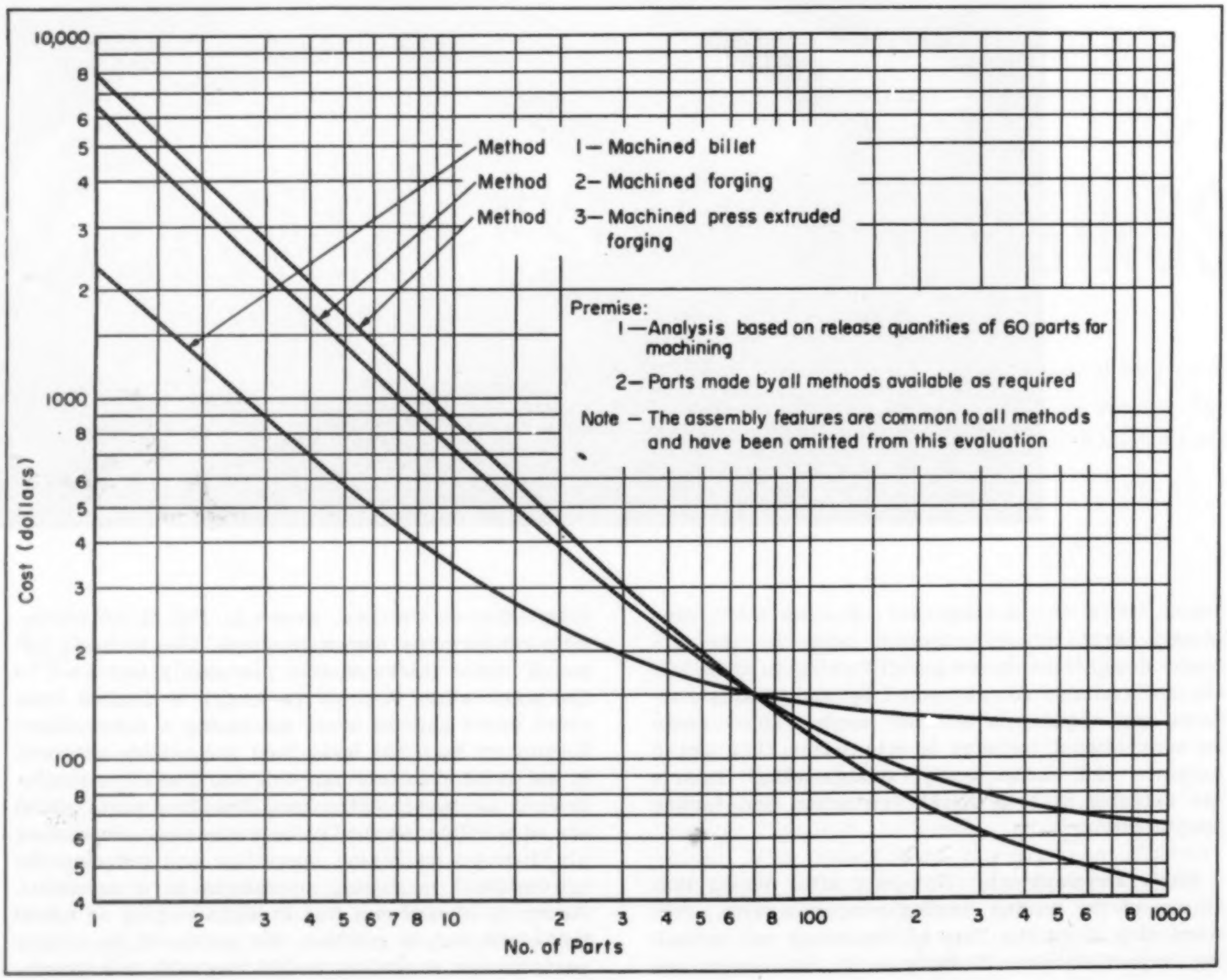


Table 1—Physical Properties of Press Extruded Forgings

Forging No.	Specimen No.	Yield Strength (psi, 0.2% offset)	Tensile Strength (psi)	Elong. (% in 2")	Hardness (Rock. B)
1	1	61,670	65,520	3.0	81–81
1	2	59,590	65,670	7.5*	79–80
1	3	62,500	69,240	10.0	81–81
1	4	60,810	67,340	7.0	80–80
1	5	61,780	68,530	7.0	80–81
1	6	57,970	66,370	7.0	79–81
1	7	59,460	66,370	7.0*	80–81
1	8	57,890	66,470	9.0	77–78
2	1	61,660	67,770	5.0	80–81
2	2	60,450	68,990	8.0*	80–82
2	3	61,710	69,030	8.0	80–82
2	4	61,860	68,730	7.0	81–81
2	5	63,580	69,940	8.0	81–82
2	6	61,110	68,160	7.0	81–81
2	7	60,030	67,420	7.0*	80–81
2	8	60,590	67,760	9.0	81–81

* Denotes 1-inch gage length.

Note: AMS-4135G minimum requirements per paragraphs 5.1.1.1 and 5.1.1.2 are as follows:

Tensile Strength (psi)	Yield Strength (psi, 0.2% offset)	Elong. (% in 2")
65,000	55,000	7.0 Longitudinal†
62,000	52,000	3.0 Transverse

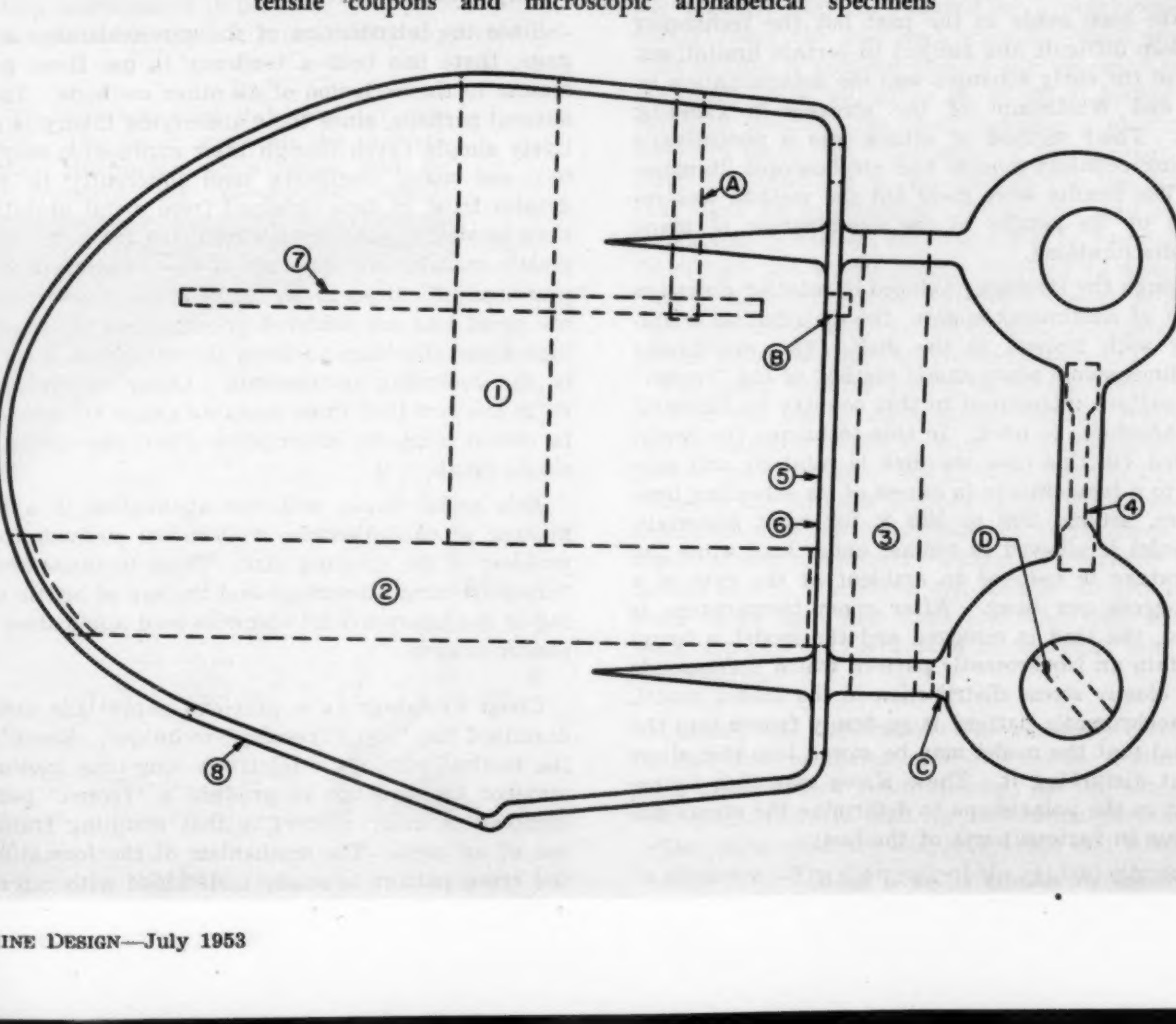
† Minimum percentage of elongation requirement when test specimens are taken from forgings that are less than four (4) inches in thickness.

to the point where several hundred acceptable parts have been made.

From the first production shipment, two parts were chosen at random and subjected to a detailed physical and metallographic evaluation. Physical test specimens and metallographic specimens were cut from the parts as shown by *Fig.* 6. Particular interest was placed in obtaining specimens from areas which were known to be troublesome during the development of the forging techniques. From each part, a total of eight physical test specimens and four metallographic samples were tested with the results as shown in TABLE 1. These results were considered to be more than satisfactory since all portions of the part exceeded minimum specification properties as called for in the 14ST Forging Specification No. AMS4135.

Grain flow studies indicated excellent metal flow control with a resulting grain direction that was oriented in a most efficient manner. Upon viewing these grain flow studies, the inefficiency of machining away good grain flow characteristics is readily apparent. As a result of experience to date, it seems relatively reasonable to predict that eventually, perhaps in the near future, it will not be impact extrusions versus forgings, but essentially a combination of the two—in a process exemplified by the aileron nose rib—that will be utilized for a very sizeable portion of present forgings.

Fig. 6—Diagrammatic sketch showing location of numerical tensile coupons and microscopic alphabetical specimens



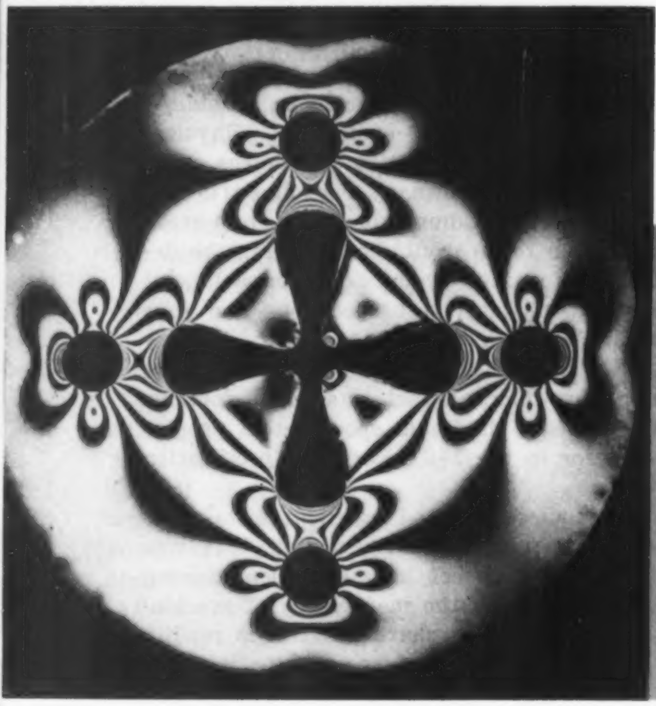


Fig. 1—Isochromatic photograph of a Marblette perforated rotating disk, obtained by the "creeping" method. The disk was driven at 5300 rpm for 19 hours at room temperature

STRESSES IN

By R. L. Lake and A. J. Durelli
Armour Research Foundation
Illinois Institute of Technology
Chicago, Ill.

THE turbo-jet engine is responsible for a renewed and more widespread interest in the problem of the distribution of stresses in bodies rotating at high speed. Several experimental attacks on this problem have been made in the past but the techniques have been difficult and subject to certain limitations.

One of the early attempts was the determination by Frost and Whitcomb[1] of the stresses in grinding wheels. Their method of attack was a photoelastic one using celluloid models and stroboscopic illumination. The results were good but the method was restricted by its nature to the examination of plane stress distributions.

Although the stresses produced in rotating disks are a result of continuous motion, the distribution is stationary with respect to the disk. The well-known three-dimensional photoelastic method of the "frozen" stress pattern introduced to this country by Hetenyi[2] may, therefore, be used. In this technique the model is loaded (in this case the disk is rotated) and subjected to a temperature in excess of its annealing temperature, around 200 to 250 F for most materials. The model is allowed to remain under load while the temperature is reduced to ambient at the rate of a few degrees per hour. After room temperature is reached, the load is removed and the model is found to contain an isochromatic pattern which corresponds to the elastic stress distribution in the loaded model. This isochromatic pattern is so firmly frozen into the material that the model may be sawed into thin slices without disturbing it. These slices may then be examined in the polariscope to determine the stress distribution in various parts of the body.

1. References are tabulated at end of article.

The method is quite satisfactory and has been applied more recently by Bernhart, Hale and Meriam.[3] Its chief disadvantage is the requirement of an elaborately controlled oven.

Since the introduction of the wire-resistance strain gage, there has been a tendency to use these gages almost to the exclusion of all other methods. This is natural perhaps, since their underlying theory is relatively simple (even though their application may not be) and many engineers tend inherently to place greater trust in data obtained from metal prototypes than in similar information resulting from the use of plastic models. In the case of the rotating disk the practical difficulties in the use of wire-resistance gages are great and are centered primarily on the need for high-speed slip rings to bring the data from the gages to the recording instruments. Other disadvantages lie in the fact that three separate gages are necessary to obtain complete information about the stress at a single point.

This article deals with the application in a novel manner of two otherwise well-known methods to the problem of the rotating disk. These methods are the "creep-freezing" technique and the use of brittle coating in conjunction with stepwise load application and plastic models.

Creep Freezing: In a previous paper[4] the authors described the "creep freezing" technique. Essentially the method employs a relatively long-time loading at ambient temperature to produce a "frozen" pattern identical in every respect to that resulting from the use of an oven. The mechanism of the formation of the creep pattern is easily understood with reference

Fig. 3—Light field detail of the Marblette "crept in" isochromatic pattern

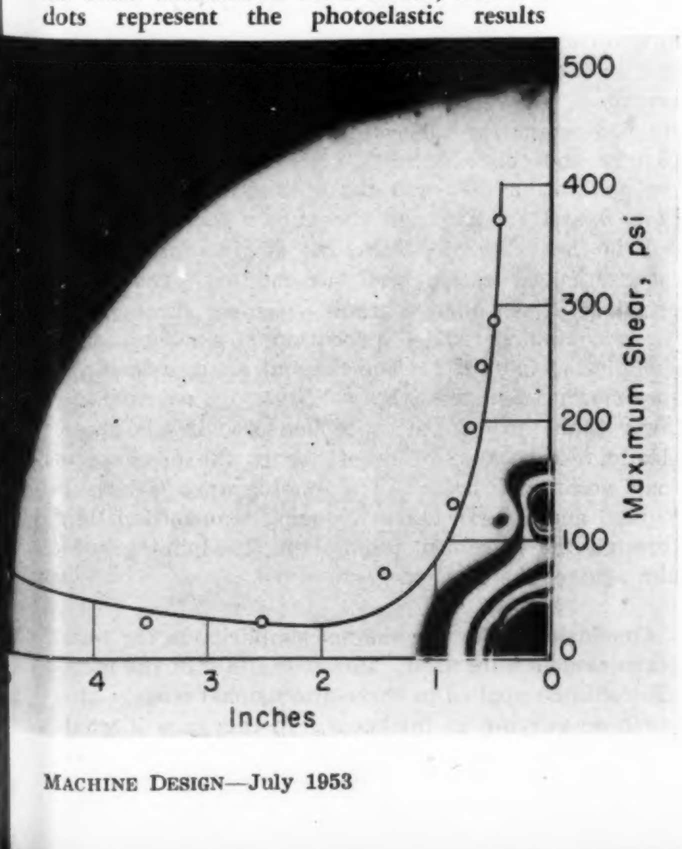
ROTATING PARTS

How two experimental techniques can be applied in a new way to stress determinations with plastic models

to the usual spring and dashpot analogy for hardening resins. This analogy which is fully explained by Hetenyi[2] likens the behavior of the plastic to that of a spring and dashpot in parallel.

When a load is applied, it is initially supported primarily by the dashpot, but as time passes the spring gradually takes over as the dashpot yields. At a sufficiently high temperature, the effective viscosity of the dashpot may be reduced to a point where the spring takes the full load at once. Then when the temperature is reduced slowly, the viscosity is again increased until the spring is, in effect, locked in the extended position. In creep freezing, a plastic is selected in which the "viscosity" of the dashpot at ambient temperatures is low enough for the spring to take over an appreciable part of the load within a reasonable period of time, say ten to twenty hours. Of course, the spring begins to relax as soon as the load is removed but the rate of recovery is so slow that the model may be sliced and photographs obtained before the pattern becomes distorted.

Fig. 2—Marblette solid disk used for calibration. The curve represents the theoretical shear distribution on a radius, and the dots represent the photoelastic results



The usual plastics employed in the creep-freezing method are phenolformaldehyde casting resins typified by Bakelite 48-306, Catalin, general purpose transparent and Marblette. All of these plastics have been tested in the authors' laboratory and it has been demonstrated that the frozen pattern corresponds to the elastic stress distribution within a reasonable experimental error. Marblette was used in the photoelastic tests to be described since the other two materials were not generally available at the time this article was written.

Loading and Calibration: The geometry of the perforated disk is shown in *Fig.* 1. *Fig.* 2 shows the solid disk, rotated simultaneously on the same shaft with the perforated disk for calibration purposes.

The disks used are 0.25-inch thick and 9.90 inches in diameter. The diameter of the central circular hole

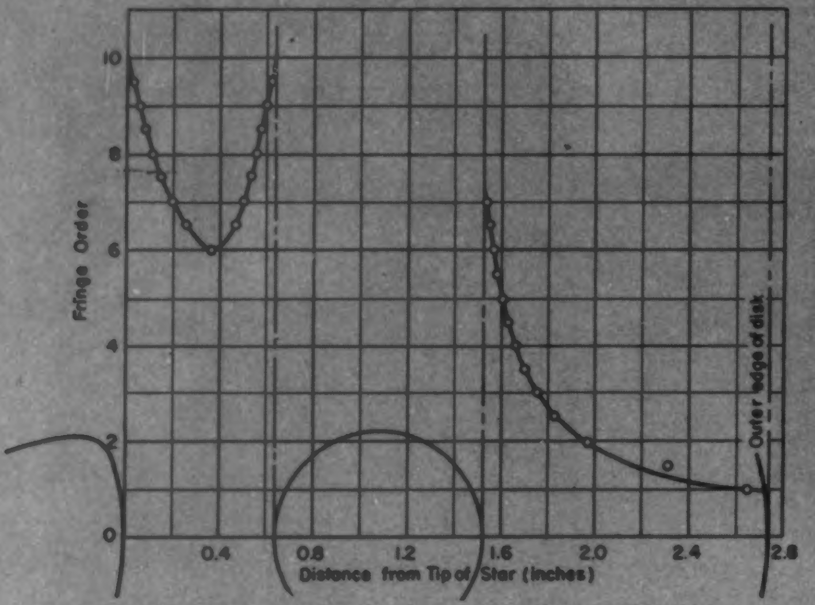


Fig. 4—Fringe order along a radius passing through the tip of the star. Speed, 5300 rpm

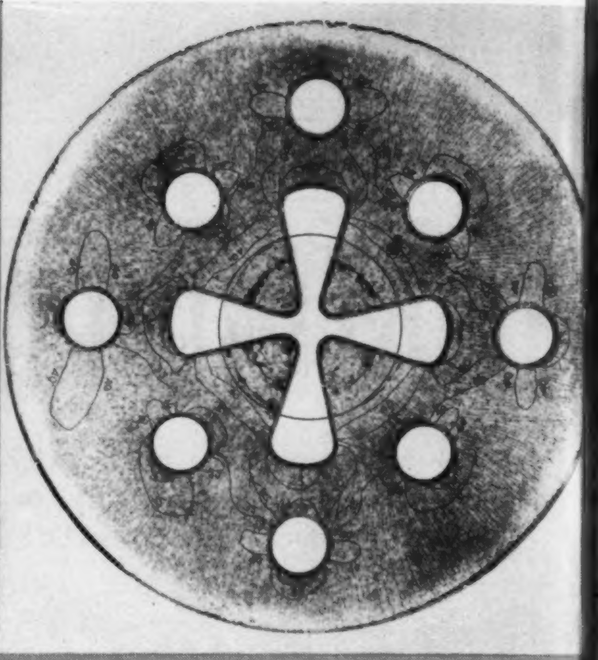


Fig. 5—Brittle coating photograph of a Catalin perforated rotating disk. The last isoentatic was obtained at 4000 rpm

of the calibration disk is 0.88-inch. The driving torque is transmitted from the shaft to the calibration disk through two screws inserted into two small holes, one of which is shown in *Fig.* 2. The star-perforated disk is driven by two screws inserted at two diametrically opposite points inside the tips of the star. After these disks had been driven at 5300 rpm for a period of 19 hours, they were taken out and studied in the diffused light polariscope.

Photoelastic Test Results: The isochromatic patterns of the calibration disk and of the specimen are shown in *Figs.* 1, 2 and 3. The formula for the inertia stresses in a rotating disk with a central circular hole was used to compute the stresses at points along the diameter of the calibration disk, *Fig.* 2. The material fringe constant could then be calculated from

$$f = \frac{\sigma_\theta - \sigma_r}{2n} t$$

where f = fringe constant, psi shear per inch of thickness; σ_θ = tangential stress, psi; σ_r = radial stress, psi; t = thickness, inches; and n = number of fringes.

An average value of f = 25 psi per inch of thickness was used in computing the theoretical curve of *Fig.* 2. The fringe order along an axis of symmetry through a branch of the perforation is plotted in *Fig.* 4. With the average fringe value of 25 psi in the formula given, σ_θ at the end of the perforation may be computed to be 2000 psi, since σ_r must vanish at the free boundary. The principal stresses within the field could be computed by any one of several methods if desired. Most of these methods would require additional information which could be obtained in the polariscope or by independent means.

Brittle Coating Test: The geometry of the second perforated disk is shown in *Fig.* 5. The model was made of Catalin, and Stresscoat was used to determine the isostatics and the larger principal stress. The calibration disk and the specimen were rotated side by side to a certain speed and stopped for inspection of cracks. They were then rotated at a higher speed and stopped again for inspection of cracks. The crack pattern and the isoentatics (loci of cracks ends) of the calibration disk and the specimen are shown in *Figs.* 6 and 7. Through the known stress distribution in the calibration disk, the strain sensitivity of the particular coating used is computed. The larger principal stress along a radius passing through the tip is computed from the location of the isoentatics and plotted in *Fig.* 8. The tangential stresses in the specimen used in the Stresscoat test are considerably lower than those in the specimen used in the photoelastic test because of relief due to the presence of four additional holes. The photographs shown in *Figs.* 6 and 7 were taken by using transmitted light through the specimen and holding the photographic film against the coating.

Conclusions: For the sake of simplicity in the tests, plane models were used. However either of the methods could be applied to three-dimensional models, that is, those varying in thickness. In this case it would

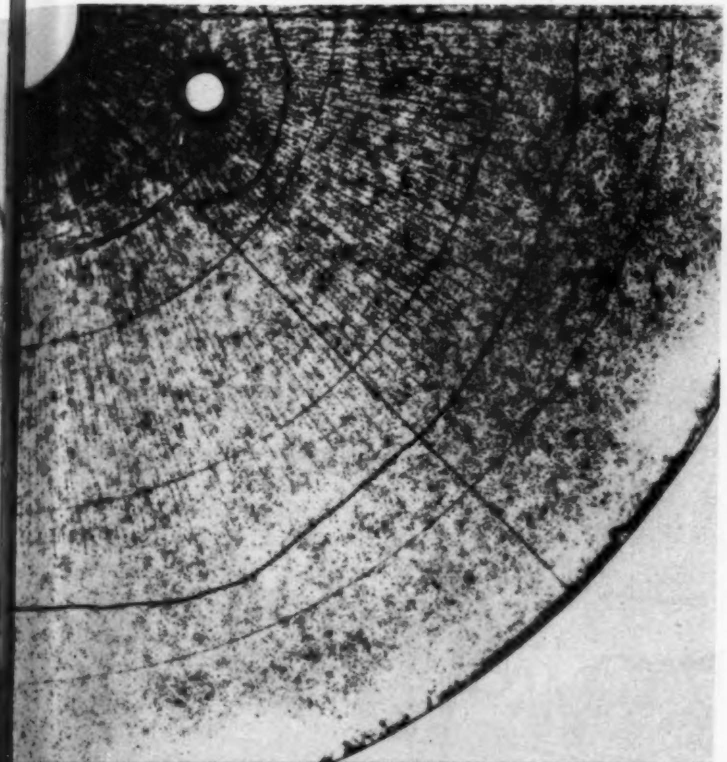
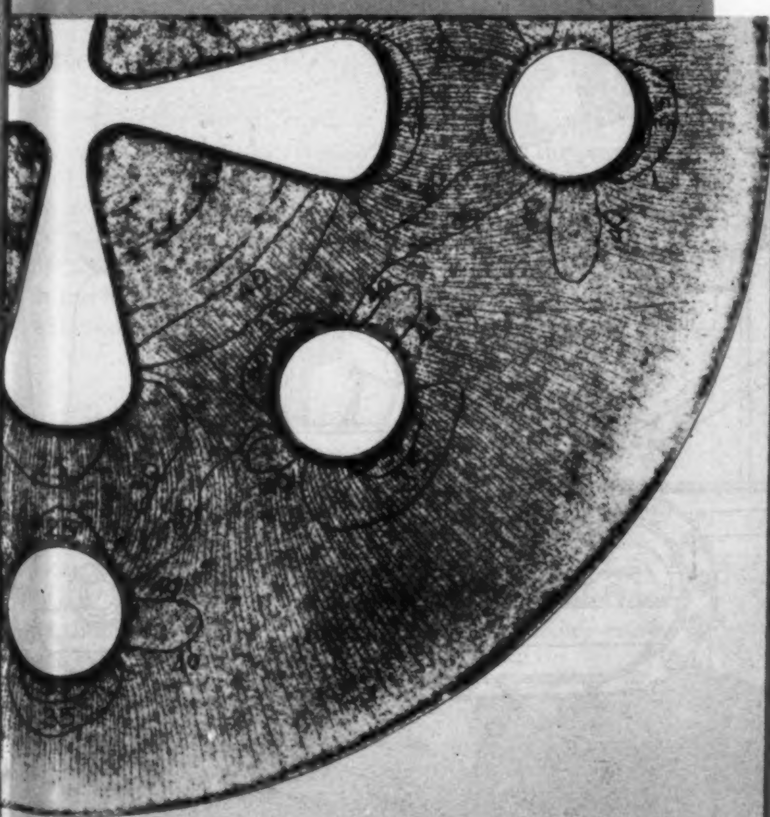


Fig. 6—Above—Calibrating disk for the brittle coating test. Numeral 40 on isoentatic corresponds to 4000 rpm

Fig. 7—Below—Brittle coating stress determination on a Catalin rotating disk. The thin lines are cracks and coincide with the principal stress trajectories. The heavy lines are isoentatics and have been drawn after each increment of speed



be necessary to slice the photoelastic model to obtain an accurate picture of the stress distribution. Even though the axial stresses were not expected to be of great magnitude, it is probable that the other two principal stresses would not remain in an axial cross-sectional plane. Moreover, this rotation would influence the final integrated effect if the complete model were viewed in the polariscope.

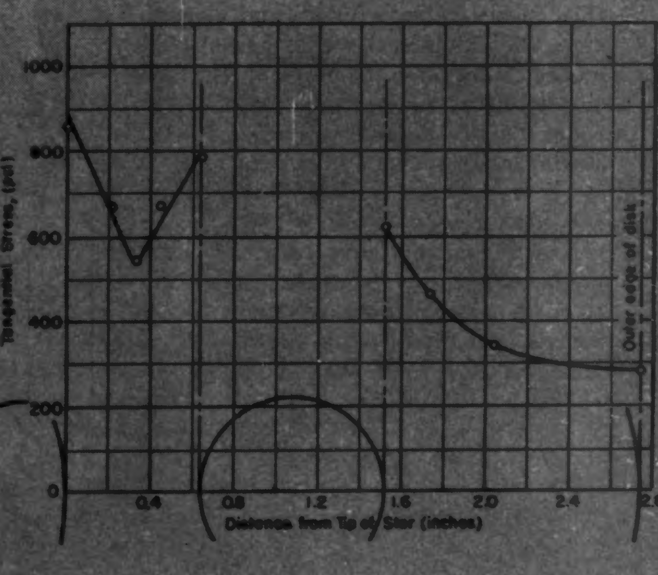
The brittle-coating test, on the other hand, would be conducted in exactly the same manner regardless of the shape of the model. The only question which must be decided is whether the metal prototype should be used with the difficulties inherent in building up and braking down to the high speeds required to produce appreciable strain, or whether a plastic model should be manufactured for testing at much lower speeds.

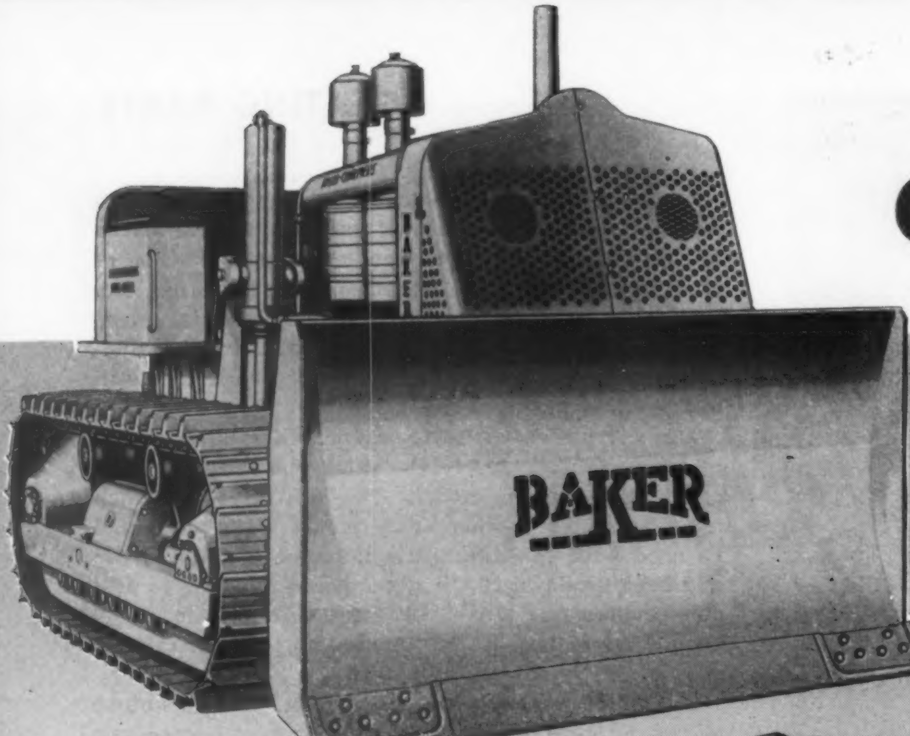
As might be expected, the photoelastic technique allows a more precise and more complete analysis than does the brittle coating. However, the coating test may be run very rapidly and will yield information on the surface stresses which in most cases are the important ones.

References

1. T. H. Frost and K. F. Whitcomb—"Stresses in Rotating Disks," *Trans. ASME*, Vol. 53, No. 3, 1931.
2. M. Hetenyi—"The Application of Hardening Resins in Three-dimensional Photoelastic Studies," *J. Appl. Physics*, Vol. 10, 1939, Pages 295-300.
3. E. E. Bernhart, A. L. Hale and J. L. Meriam, "Stresses in Rotating Disks Due to Noncentral Holes," *Proc. SESA*, Vol. 9, No. 1, Pages 35-52.
4. A. J. Durelli and R. L. Lake—"Some Unorthodox Procedures in Photoelasticity," *Proc. SESA*, Vol. 9, No. 1, Pages 97-122.

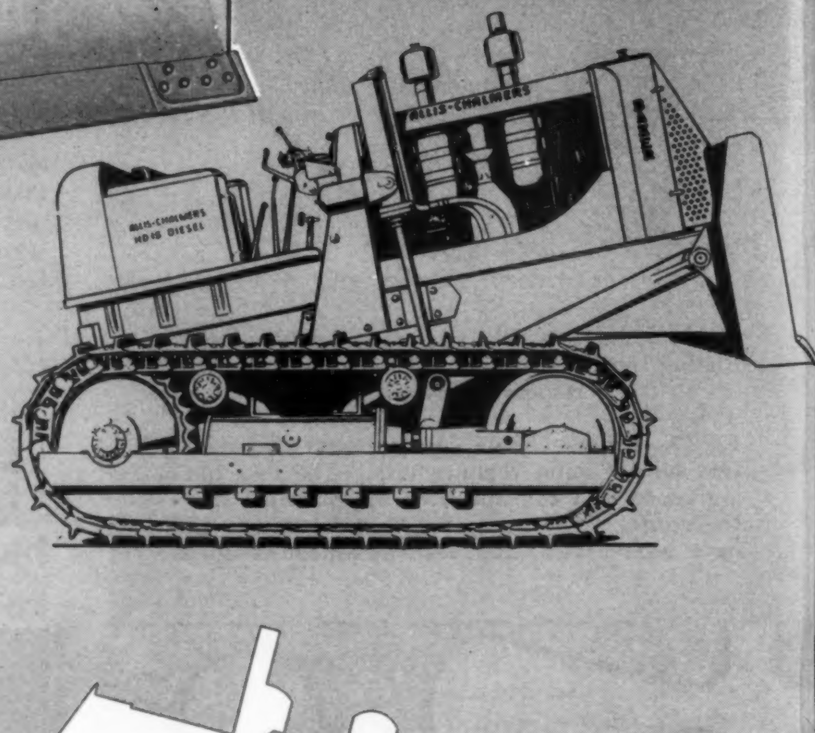
Fig. 8—Tangential stresses at points along a radius passing through the tip of a star, determined by brittle coating method. Speed, 5300 rpm





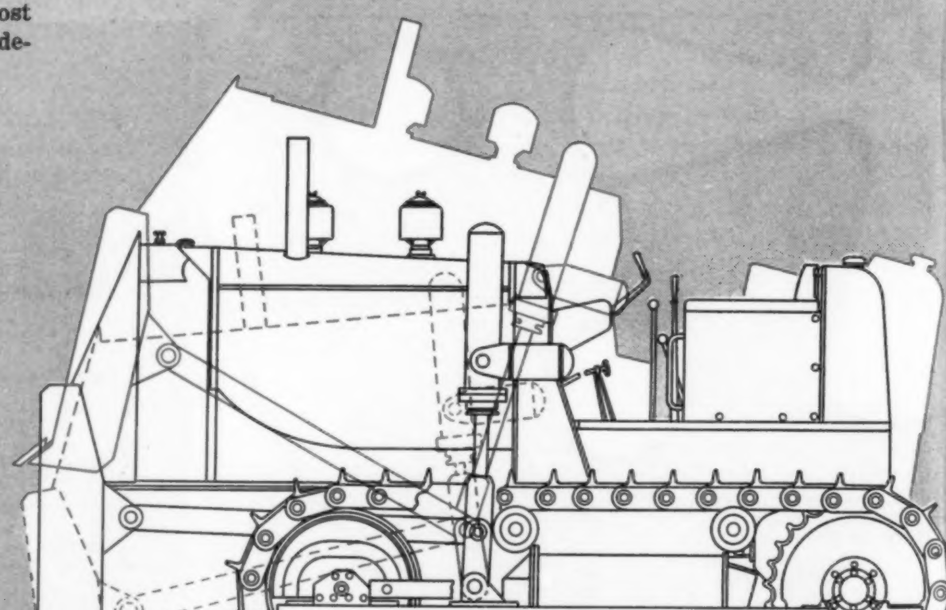
CONTEM

Hydraulically Sprung Frame Eliminates Bulldozer Pushbeam



THE ENTIRE front tractor frame and integrally bolted dozer attachment are supported by twin hydraulic cylinders on a new bulldozer, eliminating the conventional pushbeam. Designed by the Baker Mfg. Co., after development of a smaller pilot model last year, the new 15X bulldozer is mounted on a modified Allis-Chalmers 109-horsepower crawler-tractor. This simplified design has permitted a reduction in dozer blade weight to 5366 pounds, almost 1400 pounds less than a conventional design would weigh.

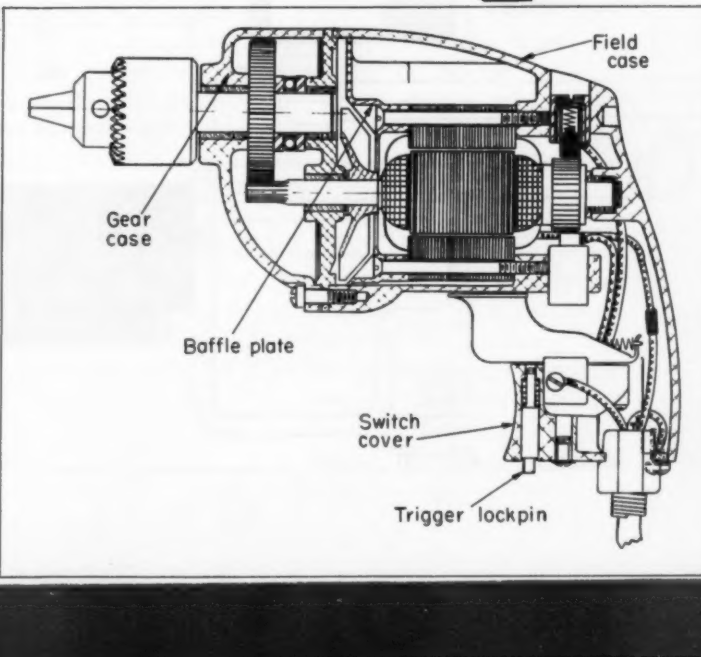
The tractor main spring is removed, permitting the entire frame and dozer to be tilted to provide a blade bite of 15½ inches below ground, and a maximum blade height above ground of 39½ inches. Stabilizer bars from the hydraulic piston ends to the moldboard help absorb torsional strain and reduce horizontal forces on the tractor frame. With the new design, more of the tractor's weight and engine horsepower can be converted into positive down pressure. Four control positions — up, down, hold and float—are available.



Integral Die-Cast Handle Simplifies Drill Design

A STURDY, compact design, with reduction in machining and assembly operations, is attained in the ¼-inch Copper Line drills of Thor Power Tool Co. by die casting the handle and field case in one piece. Use of aluminum alloy diecastings for field and gear cases has reduced total weight to 2¾ pounds.

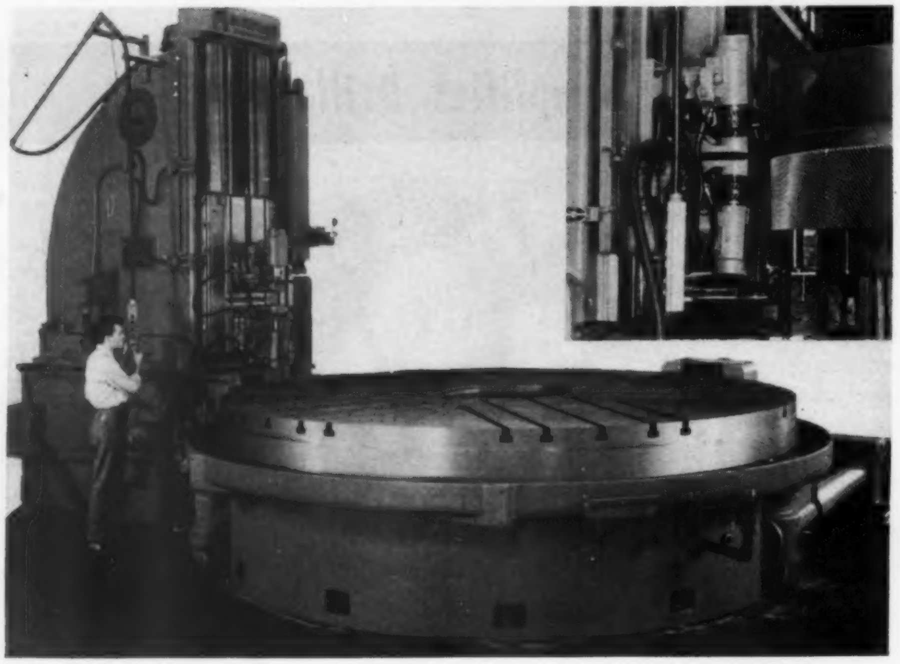
A separate switch cover is mounted with only two screws; two locating lugs at the top of the switch cover hold the cover tight to the field case without additional screws. Extra handle strength is provided by the channel shape and the reinforcement provided by screw bosses inside the handle. A die-cast hole in the bottom of the switch-cover holds a cable clamp, which acts as both a cord sleeve and strain reliever.



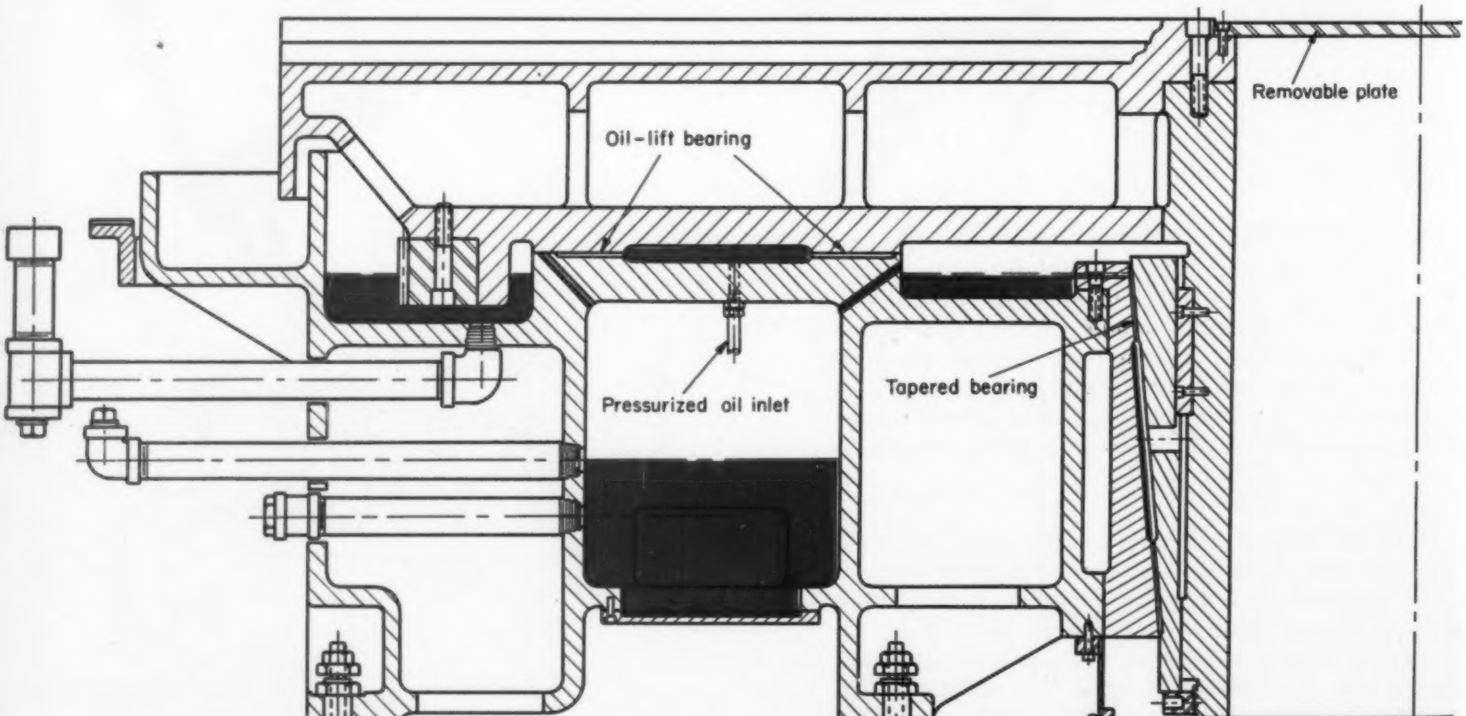
Cool operation is aided by a baffle plate which directs the ventilating air over the motor to the centrifugal fan, which is a specially designed, large-volume type. One of the main attempts in the design has been to provide a good balance between radiated and conducted heat to prevent hot spots. An additional feature of the design is the use of a trigger lockpin for continuous operation of the two-pole, momentary-contact switch.

Oil-Lift Bearing Supports Gear-Shaver Table

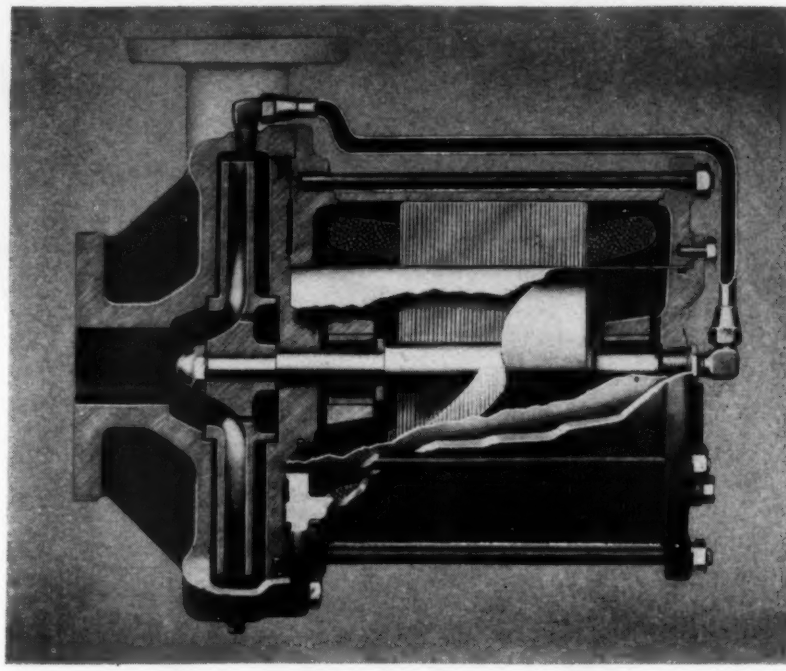
INTERNAL and external gears up to 15 feet in diameter and 65 inches wide can be handled on the new Michigan Tool Co. V-180 rotary gear shaver. A 36-inch diameter hole in the work table permits shaving gears with integral shafts. Dual cutting heads are used, independently adjustable for correct angle, and the top head may be adjusted for height.



Since the machine must support a variety of gears weighing up to several tons apiece, the table has been designed with a hydrostatic oil-lift bearing, which supports approximately 90 per cent of the table load. Proper bearing clearance is secured despite differing gear weights by varying oil pressure to match the load. Radial thrust is carried by a gravity-fed tapered bearing. This method of support makes accuracy of several ten-thousandths of an inch possible, even on large gears.



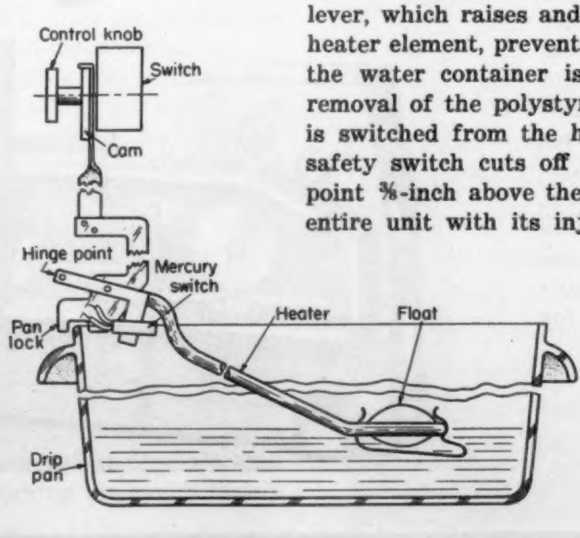
Pump Operates Without Seals or Stuffing Boxes



THE COMBINATION rotor-impeller of Chempump Corp.'s new centrifugal pump is completely enclosed in a nonmagnetic alloy sheath, eliminating all dynamic seals or stuffing boxes. The fluid being pumped circulates freely through the rotor chamber of the motor, which is isolated from stator windings. Radioactive, highly volatile, explosive, corrosive, toxic or other hazardous fluids can thus be handled safely. To prevent temperature buildup, a recirculation line feeds approximately 0.5 per cent of the total fluid pumped to the rear plug of the rotor chamber for heat dissipation. Graphitar bearings are used, so that no lubrication is required.

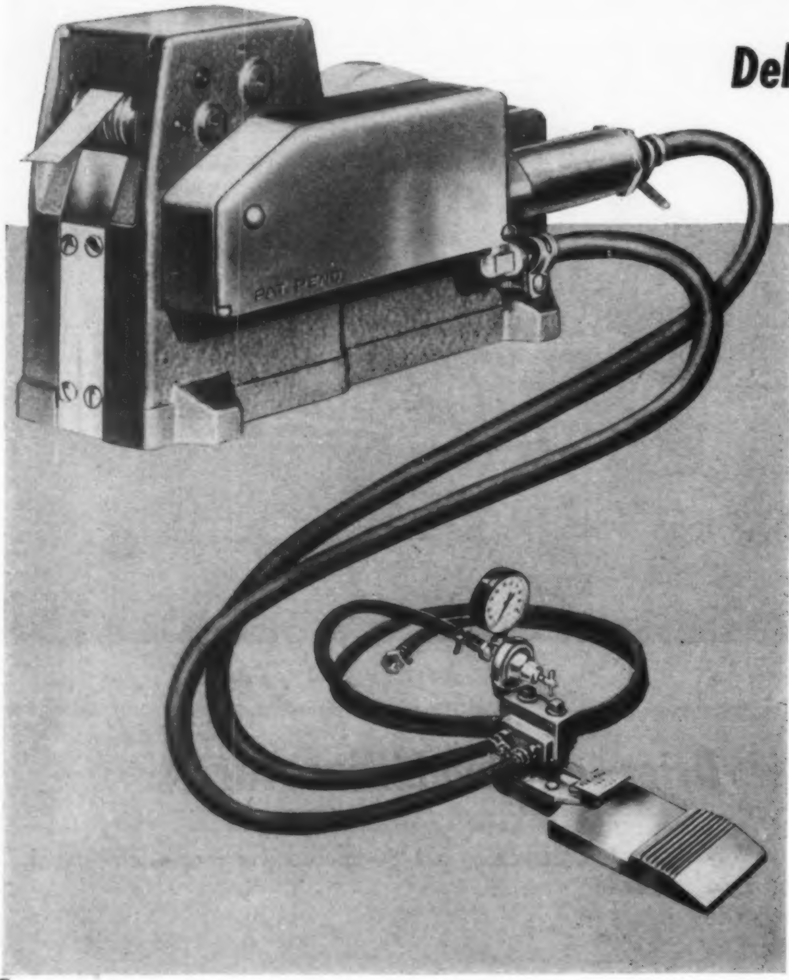
Moisture Conditioner Dries or Humidifies Air

A REFRIGERATING-type humidifier and a vaporizing humidifier are both included in the new Admiral Moisture-Conditioner. As a dehumidifier, the unit removes up to 3 gallons of moisture every day; as a humidifier, it adds up to 2 gallons. The dehumidifier refrigeration system is of the capillary type, consisting of a Tecumseh ⅛-horsepower compressor, 8 feet of Wolverine Tru-Fin tubing formed into a three-turn coil, and a brazed aluminum tube-on-sheet evaporator plate. Freon-12 is the refrigerant. A separate propeller fan, driven by a 1/80-horsepower motor, circulates the air.

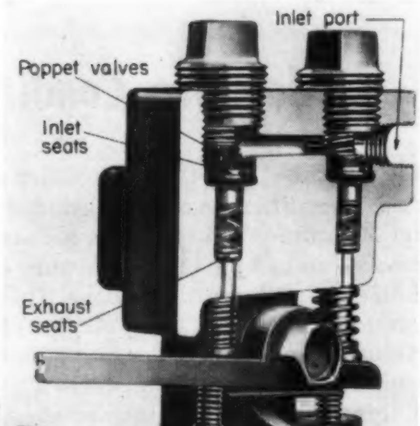


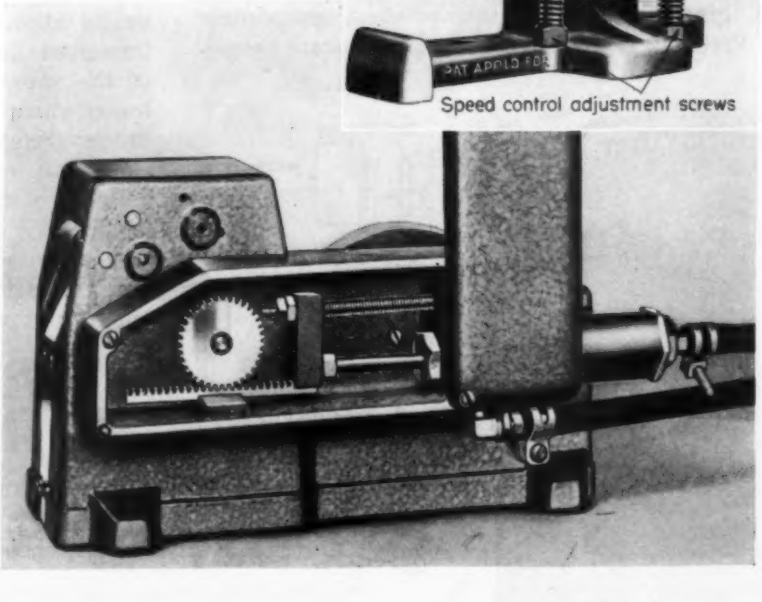
When the control knob is turned to the "humidify" position a 250-watt heater is lowered into a 3-gallon container of water by a cam on the control-knob shaft and associated linkage. A float attached to the heater allows the "live" section of the heater to be immersed approximately ⅜-inch under the surface of the water. As a safety feature, an interlocking lever, which raises and lowers with movement of the heater element, prevents lowering of the heater unless the water container is in place, and also prevents removal of the polystyrene plastic pan until the unit is switched from the humidifying cycle. A mercury safety switch cuts off the heater when it reaches a point ⅜-inch above the bottom of the container. The entire unit with its injection-molded plastic housing weighs 52 pounds, is less than 2 feet high, and is about 16 inches in diameter.

Air-Operated Tape Dispenser Delivers Metered Lengths

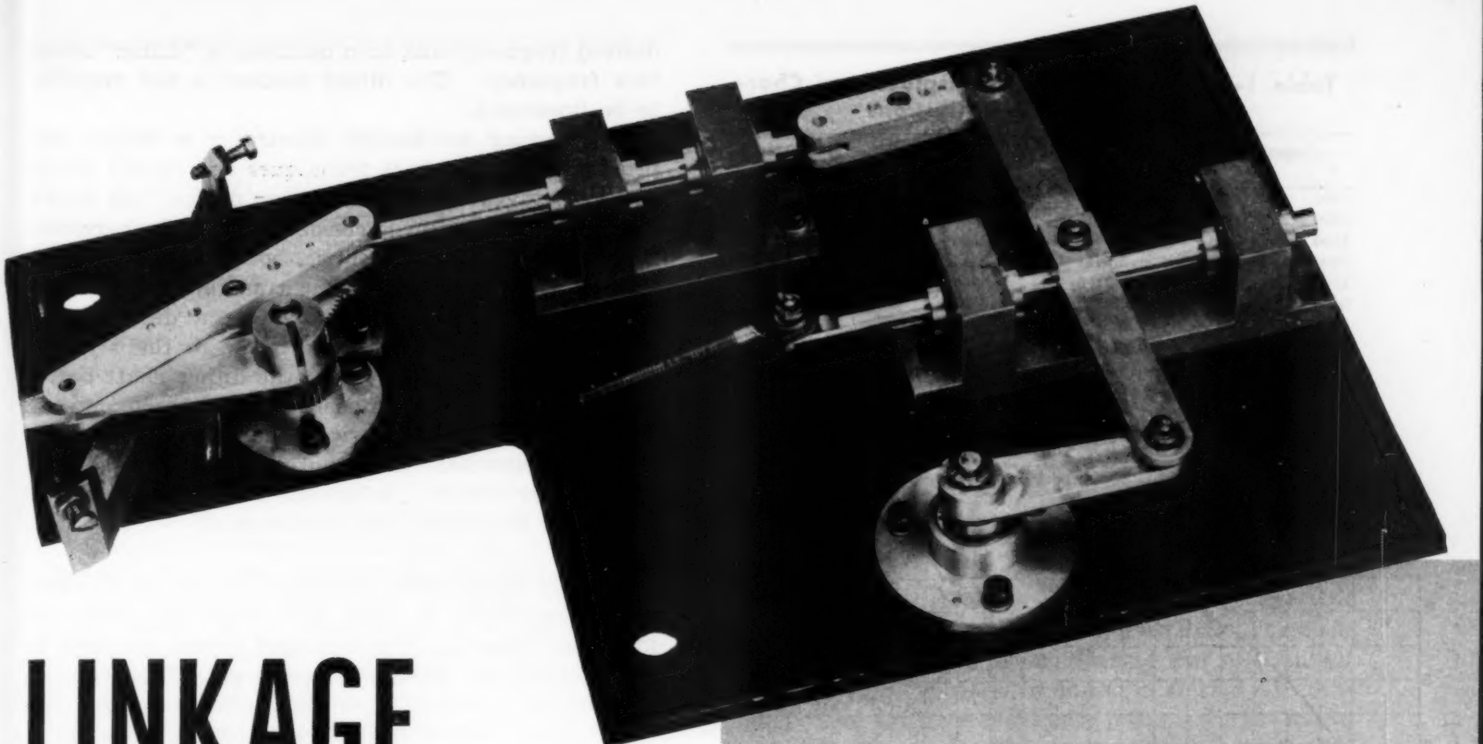


A PREDETERMINED length of pressure-sensitive tape is delivered by the model AF-92 dispenser at the touch of a foot valve. Designed by Air Fixtures Inc., the unit handles tape rolls from ¼ to 1-inch wide; a larger unit is also available for tape to 2 inches wide.





Tape feed is through a set of rollers driven from a double-acting cylinder through a rack and gear. An over-running clutch permits the rollers to stay in a neutral position as the rack is returned. Length of tape is controlled by a stroke adjustment screw which determines the starting point of each stroke. A speed control is also incorporated in the foot valve for adjusting stroke speed.

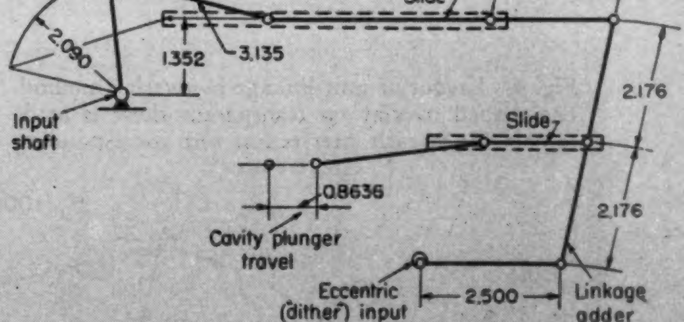


LINKAGE DESIGN

By Philip T. Nickson
Radar and Communications Div.
Raytheon Manufacturing Co.
Waltham, Mass.

Fig. 1—Above — Cavity-tuning mechanism used with a klystron illustrates application of overlay method in design of linkages

Fig. 2—Below—Arrangement of linkage for klystron cavity tuner. Frequency output of the cavity must be in linear proportion to the rotation of the input shaft



A mechanism for giving a desired input-output relationship is derived by the overlay method. Applied here to a klystron tuning cavity mechanism, the technique is applicable to a wide variety of motion relationships. Means for providing adjustability are explained, and the mechanical details of joints and sliders of high accuracy are described and illustrated

IN A PREVIOUS article[1] a systematic procedure for linkage layout was explained. The present article considers an actual linkage mechanism designed by this procedure, and shows some of the design features of the device, which is illustrated in *Fig.* 1. A method is described for making the mechanism adjustable, thereby greatly enhancing its practicability. For the most part the conventions of the previous article are followed.

This mechanism might simply be called an input-output device, which in this instance drives the plunger of a klystron tuning cavity. The klystron is used as the local oscillator in a radar set, and it is required that the frequency-displacement characteristics of the cavity be linear. In other words, considering the linkage input to be shaft rotation through a limited angle, the frequency output of the cavity must be in linear proportion to shaft travel. Also it is required that the mechanism be capable of tuning the cavity to any

1. References are listed at end of article.

Table 1—Average Frequency-Displacement Characteristics of Eight Klystrons

—Input—		——Output——		Deviation	
f	f_n*	d_{ave}	d_n†	ϵ	ϵ'
1000	0	0.9550	0.0000	0.00	0.00
1200	0.1	0.8471	0.1365	3.65	3.43
1400	0.2	0.7406	0.2598	5.98	6.00
1600	0.3	0.6417	0.3744	7.44	7.63
1800	0.4	0.5450	0.4863	8.63	8.43
2000	0.5	0.4608	0.5838	8.38	8.47
2200	0.6	0.3826	0.6744	7.44	7.75
2400	0.7	0.3038	0.7656	6.56	6.48
2600	0.8	0.2341	0.8463	4.63	4.73
2800	0.9	0.1660	0.9252	2.52	2.52
3000	1.0	0.1014	1.0000	0.00	0.00

* Normalized value of f. Example: (1600 — 1000)/(3000 — 1000) = 0.3.
† Normalized value of d. Example: (0.6417 — 0.9650)/(0.1014 — 0.9650) = 0.3744.

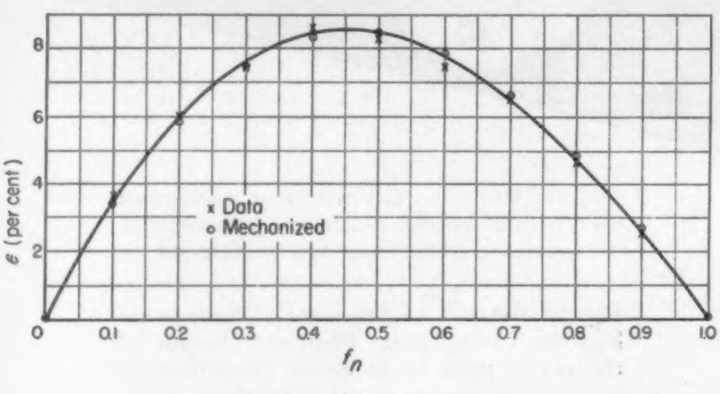


Fig. 3—Deviation of plunger displacement from linear relationship with frequency, average of eight klystrons. See last two columns of Table 1

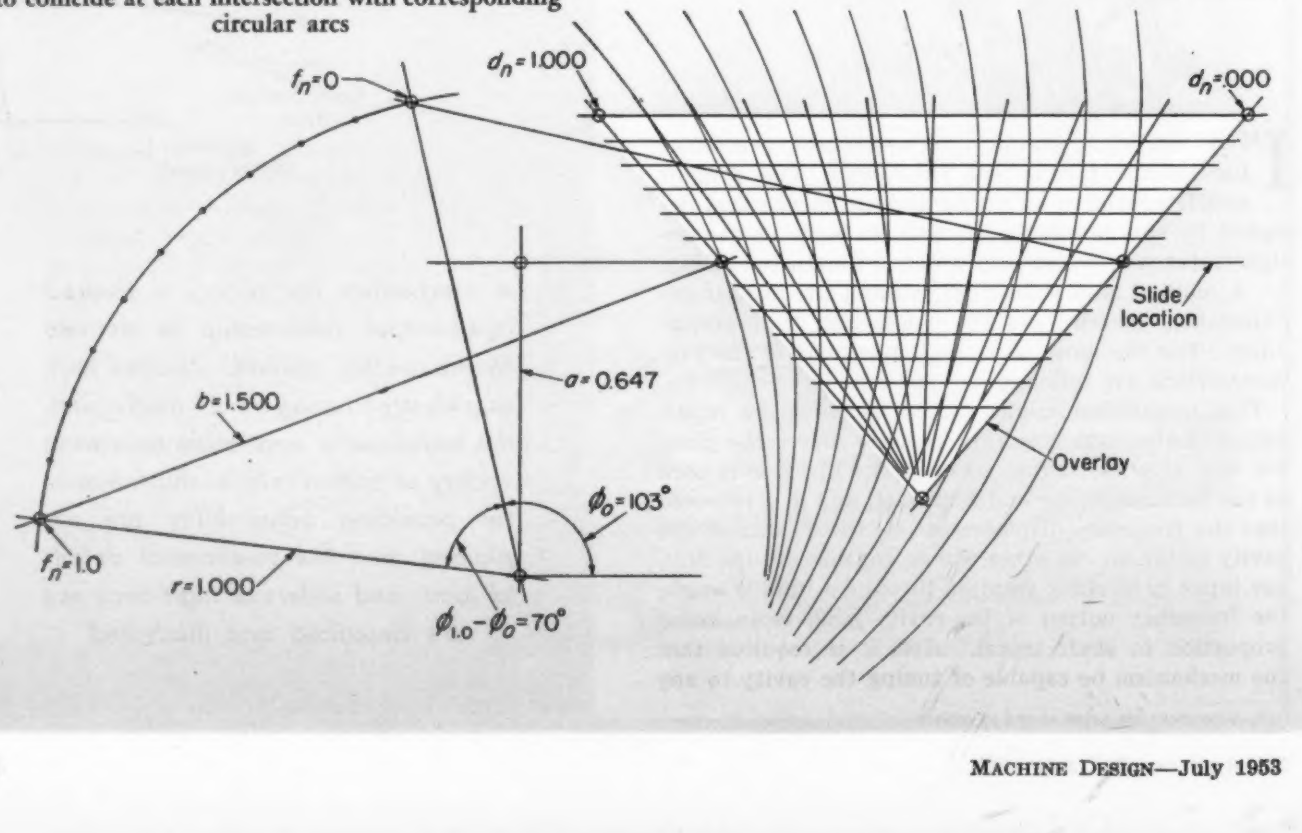
desired frequency and then oscillate or "dither" about this frequency. The dither motion is not required to be linearized.

This tuning mechanism illustrates a simple and practical use of linkage techniques and clearly shows the advantages over a cam-driven device. As shown in *Fig.* 2, the mechanism consists simply of a harmonic transformer[1] and an eccentric, both of which drive a linkage adder. The harmonic transformer linearizes the motion, the eccentric providing a dither input. The adder output is linked directly to the klystron plunger. It is necessary that the dither shaft be returned to an initial zero position when the plunger is not being oscillated, otherwise the output of the harmonic transformer would not be synchronized with the plunger travel. A special braking system was devised to accomplish this requirement.

Linkage Proportions: Frequency-displacement characteristics based on data from eight klystrons are given in TABLE 1. The averaged values are used in determining the linkage proportions. Frequency is denoted by f and plunger displacement by d. The "normalized" values are indicated by the subscript n. Deviation of d_n from straight-line relation to f_n, in per cent, is shown by ϵ. In order to smooth out the data, the deviation is plotted in *Fig.* 3. Values read from this curve appear as ϵ' in TABLE 1. These smoothed values of the deviation are used in laying out the linkage. If the data are scattered, this smoothing procedure becomes important.

The overlay method[1, 2] is used in laying out the harmonic transformer. This method is graphical and starts from estimated values of all but two of the

Fig. 4—Layout of unit linkage by overlay method. Fan-shaped overlay on transparent sheet is made to coincide at each intersection with corresponding circular arcs



linkage parameters, which are related by the expression:

$$c = r \cos \phi + [b^2 - (a - r \sin \phi)^2]^{1/2} \quad \text{..........(1)}$$

where c = output (slide) displacement, r = crank radius, ϕ = input (crank) position, a = vertical height of slide, and b = length of connecting link. The two parameters to be determined are a and the initial value of ϕ. The layout is shown in *Fig.* 4. For convenience the crank radius is taken as unity. This linkage is referred to as the "unit linkage."

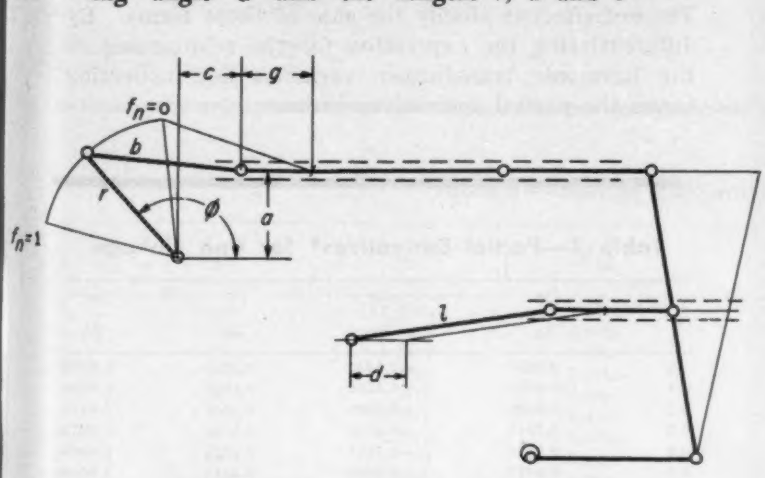
The procedure of the overlay method is as follows:

1. Assume a value of the ϕ range. For this application where the function is well behaved, i.e., where the deviation curve is smooth and fairly symmetrical, a value of 70 degrees is assumed. Experience has shown this to be a workable figure.
2. Lay out a sector with a radius of unity and an included angle of 70 degrees. The actual length of the radius on the layout drawing is taken as 5 inches. From experience again, this has been found to be a convenient scale. The reproduction in *Fig.* 4 is somewhat smaller.
3. Lay off points on the arc according to the normalized input f_n, which in this case is equal to ten equal increments of 7 degrees each.
4. Assume a value of b equal to 1.500 or 7.5 inches in length. The value of b has been found to be quite arbitrary. Any value between 1.0 to 2.0 probably would be satisfactory.
5. From each point on the arc of the sector trace an arc of radius equal to b.
6. On a separate transparent sheet, lay out the overlay. As seen in *Fig.* 4, the overlay is simply a series of parallel lines, varying in length, but each proportioned off into ten divisions according to the deviation ε' or, more correctly, according to d'_n (where $d'_n = f_n + \varepsilon'$). The length of the longest line is equal to 10 inches.
7. Place the overlay over the traces of the radius b and position it so that all eleven points on some one line of the overlay intersect corresponding traces. This position then establishes the location of the slide output of the unit linkage. The two remaining parameters are then scaled-off as: $a = 0.647$ and $\phi_o = 103.0$ degrees. If a close fit is not obtained for all intersections, it is necessary to vary either or both the ϕ range and b and to repeat the layout. Usually several trials are required. However, as in this illustration where the function is well behaved, one trial is often sufficient.

From the now determined parameters of the unit linkage, the output is calculated, Table 2. The positional error is shown by comparing the mechanized deviation, ϵ_{mech}, with the required deviation, ϵ'. Applying "zero set," the maximum point error is seen to equal 0.15 per cent. The usual accuracy to be expected from this overlay method is on the order of 0.3 to 0.5 per cent. However, with good drawing technique and a little persistence it is not unusual to achieve smaller values.

From Table 1, it is seen that the required travel of the klystron plunger is equal to $0.9650 - 0.1014 = 0.8636$-inch. The linkage adder has a 1 to 1 ratio. The linkage input then must have a range equal to twice this value or 1.7272 inches. From Table 2, the

Fig. 5—Below—Nomenclature for linkage parameters. Adjustments may be made by changing angle ϕ and the lengths r, a and b



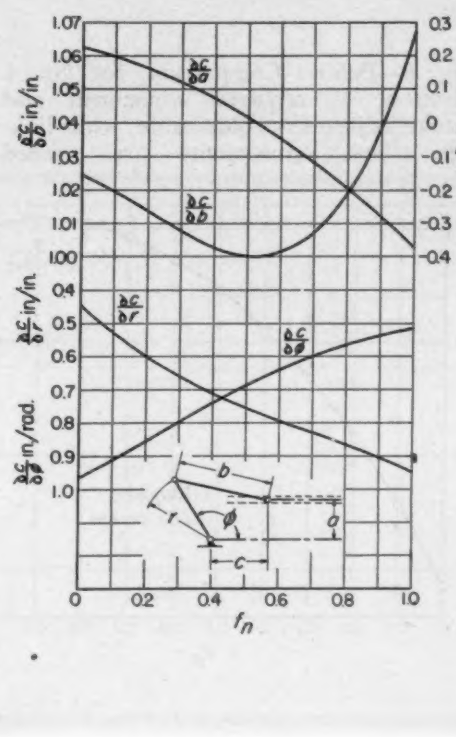


Fig. 6—Right—Plot of partial derivatives calculated from Equations 3, 4, 5 and 6. Numerical values are given in Table 3

Table 2—Calculation of Unit Linkage

f_n	ϕ	c^*	$g^\dagger$	g_n	ε_{mech}	ε'	Error	Zero Set
0	103	1.23889	0.00000	0.0000	0.00	0.00	0.00	0.10
0.1	110	1.12915	0.10974	0.1328	3.28	3.43	−0.15	−0.05
0.2	117	1.02603	0.21286	0.2576	5.76	6.00	−0.24	−0.14
0.3	124	0.92972	0.30917	0.3741	7.41	7.63	−0.22	−0.12
0.4	131	0.84007	0.39882	0.4826	8.26	8.43	−0.17	−0.07
0.5	138	0.75670	0.48219	0.5835	8.35	8.47	−0.12	−0.02
0.6	145	0.67905	0.55984	0.6775	7.75	7.75	0.00	0.10
0.7	152	0.60651	0.63238	0.7653	6.53	6.48	0.05	0.15
0.8	159	0.53839	0.70050	0.8477	4.77	4.73	0.04	0.14
0.9	166	0.47397	0.76492	0.9256	2.56	2.52	0.04	0.14
1.0	173	0.41253	0.82636	1.0000	0.00	0.00	0.00	0.10

* Calculated from Equation 1, $r = 1.000$, $a = 0.647$, $b = 1.500$.
† See *Fig.* 5: $g = 1.23889 - c$.

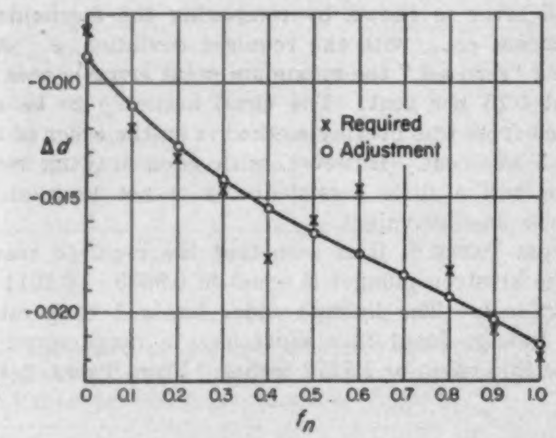
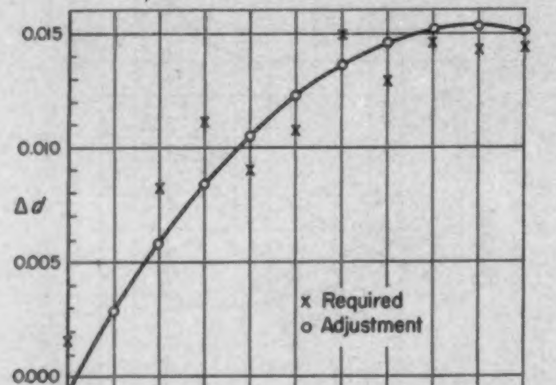


Fig. 7—Above — Comparison, for No. 8 klystron, of required adjustment and actual adjustment obtainable with linkage. Only one adjustment is needed

Fig. 8—Below—Comparison, for No. 6 klystron, of required adjustment and actual adjustment obtainable with linkage. Two adjustments are needed



travel of the unit linkage, g, is equal to 0.8264. By multiplying the parameters of the unit linkage by the ratio 1.7272/0.8264, the actual sizes in inches are obtained. They become $r = 2.090$ inches, $a = 1.352$ inches, and $b = 3.135$ inches.

Adjustment Provision: The positional error of 0.15 per cent applies only to the average plunger displacement characteristics. For any one of the eight individual cavities, from which the average is taken, the error is greater. If desired, this error can be reduced by making the linkage adjustable. Referring to *Fig.* 5, the following is an expression for making this correction:

$$\Delta d = -\frac{1}{2}\Delta\phi\frac{\partial c}{\partial\phi} - \frac{1}{2}\Delta r\frac{\partial c}{\partial r} - \frac{1}{2}\Delta a\frac{\partial c}{\partial a} - \frac{1}{2}\Delta b\frac{\partial c}{\partial b} + \Delta l\frac{\partial d}{\partial l} \qquad (2)$$

where Δd is the required adjustment and $\Delta\phi$, Δr, Δa, Δb, and Δl are the adjustments in the respective linkage parameters. Each term in the expression represents the effect of changing one linkage parameter. The net effect is simply the sum of these terms. By differentiating the expression for the relationship of the harmonic transformer variables and collecting terms the partial derivatives become:

Table 3—Partial Derivatives* for Unit Linkage

f_n	$\frac{\partial c}{\partial\phi}$	$\frac{\partial c}{\partial r}$	$\frac{\partial c}{\partial a}$	$\frac{\partial c}{\partial b}$
0	−0.9658	−0.4430	0.2237	1.0247
0.1	−0.9109	−0.5290	0.1990	1.0196
0.2	−0.8529	−0.6009	0.1649	1.0135
0.3	−0.7949	−0.6606	0.1222	1.0074
0.4	−0.7393	−0.7104	0.0720	1.0026
0.5	−0.6878	−0.7530	0.0147	1.0001
0.6	−0.6414	−0.7910	−0.0490	1.0012
0.7	−0.6006	−0.8269	−0.1192	1.0070
0.8	−0.5658	−0.8632	−0.1960	1.0190
0.9	−0.5372	−0.9024	−0.2805	1.0386
1.0	−0.5151	−0.9469	−0.2737	1.0675

*Calculated from Equations 3, 4, 5, and 6.

$$\frac{\partial c}{\partial \phi} = \frac{c\,r \sin\phi - a\,r \cos\phi}{c - r\cos\phi} \quad (3)$$

$$\frac{\partial c}{\partial r} = \frac{c \cos\phi + a \sin\phi - r}{c - r\cos\phi} \quad (4)$$

$$\frac{\partial c}{\partial a} = -\frac{a - r\sin\phi}{c - r\cos\phi} \quad (5)$$

$$\frac{\partial c}{\partial b} = \frac{b}{c - r\cos\phi} \quad (6)$$

The term $\Delta l\, \partial d/\partial l$ in Equation 2 is a constant for any one adjustment, and may be considered as a phase relationship between the outputs of the harmonic transformer and the linkage adder. The other four terms apply to the harmonic transformer and are basically the only ones concerned in the adjustment procedure.

The procedure is further explained by two examples: First, the four partial derivatives are evaluated in TABLE 3 and plotted in *Fig*. 6. In TABLE 4 an adjustment is made for cavity No. 8, for which the actual displacement is d_8. Comparing its displacement with the average displacement gives the required adjustment Δd_{req}. These values are plotted in *Fig*. 7. It is seen from a comparison with *Fig*. 6 that its shape compares generally with that of the $\partial c/\partial r$ curve. It was found that a value of $\Delta r = -0.0496$ and $\Delta l\, \partial d/\partial l = 0.0021$ resulted in values of Δd very close to those required. The linkage adjustment then is made by decreasing the crank radius, by 0.0496 inches, to 2.040 inches (2.090 − 0.050). A similar adjustment for cavity No. 6 appears in TABLE 5. Here, two adjustments in the linkage are needed. The closeness of agreement between the required and actual adjustment is illustrated in *Fig*. 8.

This method is more or less a trial-and-error process. It consists basically of summing several curves to obtain the characteristics of a desired correction. The desired correction is the plot of Δd. If the shape approximates one of the derivatives, as in the example of cavity No. 8, the adjustment can be made simply by varying only one parameter. If not, two parameters, as in the case of cavity No. 6, must be varied. Rarely do three or more parameters require adjustment. Such cases would occur only when the data, from which the average displacement characteristics are taken, are widely scattered. Here, the corrections might be beyond the scope of this adjustment procedure; that is, the required adjustment of a parameter

Table 4—Linkage Adjustment for No. 8 Cavity

f_n	d_{ave}	d_8	Δd_{req}	$-\frac{\Delta r}{2}\frac{\partial c^*}{\partial r}$	$\Delta l \frac{\partial d}{\partial l}$	$\Delta d^\dagger$
0	0.9650	0.9573	−0.0077	−0.0110	0.0021	−0.0089
0.1	0.8503	0.8378	−0.0125	−0.0131	0.0021	−0.0110
0.2	0.7425	0.7292	−0.0133	−0.0149	0.0021	−0.0128
0.3	0.6419	0.6273	−0.0146	−0.0164	0.0021	−0.0143
0.4	0.5482	0.5305	−0.0177	−0.0176	0.0021	−0.0155
0.5	0.4611	0.4452	−0.0159	−0.0187	0.0021	−0.0166
0.6	0.3799	0.3653	−0.0146	−0.0196	0.0021	−0.0175
0.7	0.3041	0.2858	−0.0183	−0.0205	0.0021	−0.0184
0.8	0.2329	0.2147	−0.0182	−0.0214	0.0021	−0.0193
0.9	0.1657	0.1450	−0.0207	−0.0224	0.0021	−0.0203
1.0	0.1014	0.0795	−0.0219	−0.0235	0.0021	−0.0214

*Calculated for $\Delta r = -0.0496$.
†Sum of two previous columns.

Table 5—Linkage Adjustment for Cavity No. 6

f_n	d_{ave}	d_6	Δd_{req}	$-\frac{\Delta r}{2}\frac{\partial c^*}{\partial r}$	$-\frac{\Delta b}{2}\frac{\partial c^\dagger}{\partial b}$	$\Delta l \frac{\partial d}{\partial l}$	$\Delta d^\S$
0	0.9650	0.9666	0.0016	0.0163	−0.0656	0.0486	−0.0007
0.1	0.8503	0.8495	−0.0008	0.0195	−0.0652	0.0486	0.0029
0.2	0.7425	0.7507	0.0082	0.0221	−0.0649	0.0486	0.0058
0.3	0.6419	0.6530	0.0111	0.0243	−0.0645	0.0486	0.0084
0.4	0.5482	0.5572	0.0090	0.0261	−0.0642	0.0486	0.0105
0.5	0.4611	0.4718	0.0107	0.0277	−0.0640	0.0486	0.0123
0.6	0.3799	0.3948	0.0149	0.0291	−0.0641	0.0486	0.0136
0.7	0.3041	0.3170	0.0129	0.0304	−0.0644	0.0486	0.0146
0.8	0.2329	0.2475	0.0146	0.0318	−0.0652	0.0486	0.0152
0.9	0.1657	0.1800	0.0143	0.0332	−0.0665	0.0486	0.0153
1.0	0.1014	0.1158	0.0144	0.0348	−0.0683	0.0486	0.0151

*For $\Delta r = 0.0736$
†For $\Delta b = 0.1280$
§Sum of previous three columns.

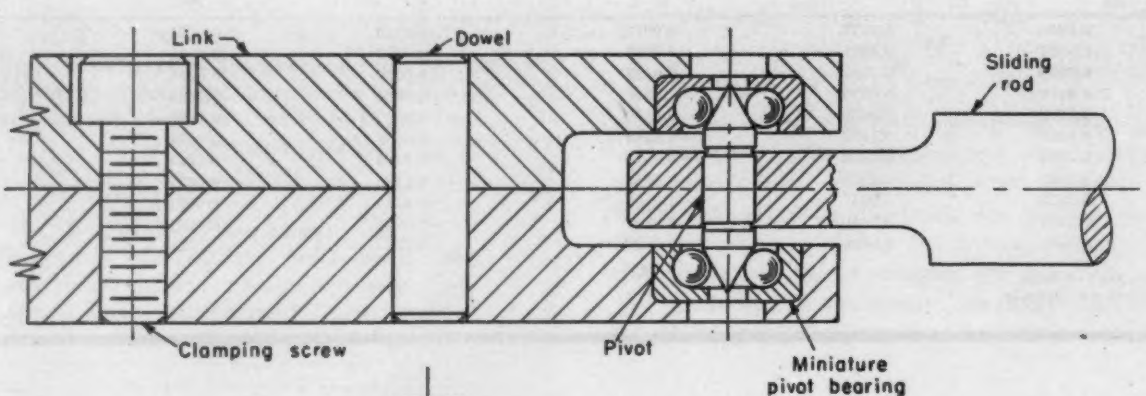
might become large in comparison to the parameter. If this is the case it would be necessary to lay out an entirely new linkage. However, the average linkage would serve as a guide in selecting the initial parameters and would make the second layout somewhat less tedious.

Linkage mechanisms consist of three basic elements: cranks, links and slides. The harmonic transformer contains one crank, one link and one slide. A double-crank (two arms 180 degrees apart), two links and three slides make up the linkage adder. In a linkage combination, one element may become common to two linkages. For example, where the harmonic transformer is used in series with the linkage adder, one slide is common. Excepting for the arrangement of the linkage, the detail design of the elements is dictated by the degree of accuracy desired. A fairly high degree of accuracy is required for this tuning mechanism. Tolerances are generally equal to ± 0.0005-inch on length dimensions and ± 0 deg 10 min on angular measurement. Minimum backlash is required on all sliding members and rotary joints.

Mechanical Construction: The actual mechanism is shown in *Fig.* 1. The aluminum baseplate is ground for flatness. All linkage parts, except shafts, bearings and hardware, are also aluminum. The two input shafts are supported on radial ball bearings. To minimize backlash, two bearings are used with each shaft and preloaded through the inner races by a bowed retaining ring. Standard ball bushings form the slides. Two bushings are carried by a bracket, which is doweled and fastened to the base plate. Hardened and ground steel rods, ¼-inch diameter, are used as the sliding members. The nominal fit between the rods and the bushings is held to 0.0003-inch.

Joints: Three types of rotary joints, using miniature pivot, radial and self-aligning bearings, appear in the mechanism. The connecting link between the crank and slide of the harmonic transformer is attached by means of a pivot-bearing joint. This joint is shown in *Fig.* 9. Two bearings of ¼-inch O.D. and 3/32-inch width are used in each joint. The link is made in two halves, which are doweled and clamped together, forming a clevis at each end. Each half of the clevis carries a bearing in a counterbored recess. A hardened-steel, conical-ended pivot pin is carried in the end of the sliding rod. The diameter of the pivot pin is equal to 0.085-inch. Each half of the link is clamped around the pivot. In the clamped position, the legs of the clevis are slightly spring loaded, thus removing backlash from the joint.

A joint using miniature radial bearings is shown in *Fig.* 10. This type of joint appears between the crank and slide of the adder and at the attachment of the adder link to the dither shaft. Two bearings, ¼-inch O.D., ⅛-inch bore and 3/32-inch width, are used in each joint. The outer races of the bearings are carried in the crank. They are separated by a spacer and are retained in the crank by spinning over the edges of the hole. A pin fastened to the sliding rod carries the inner races of the bearings. Backlash is minimized by end loading the inner races with a bowed retaining ring, acting through a shouldered washer.



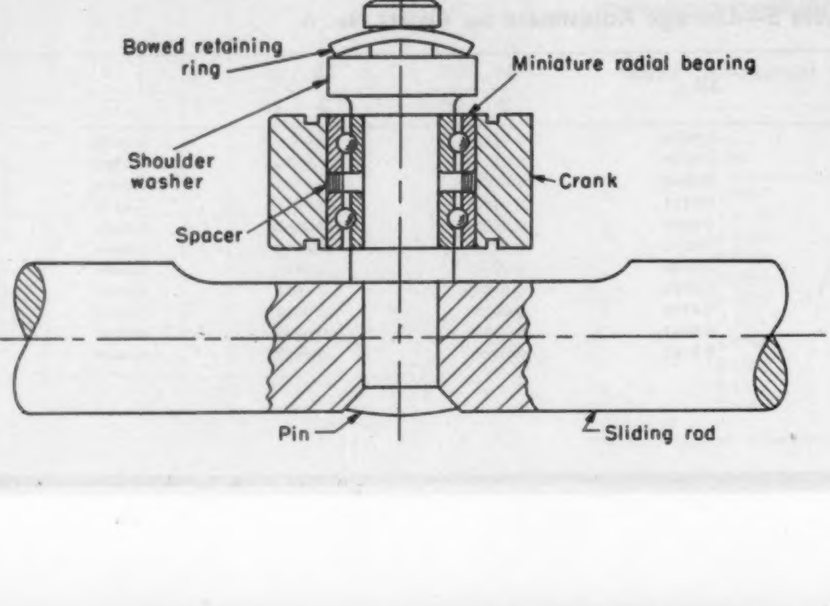


Fig. 9—Above — Detail of pivot bearing joint employed at each end of link connecting crank and slider of harmonic transformer

Fig. 10—Left — Detail of radial bearing joint used between crank and slide of the adder and at the attachment of the adder link to the dither shaft

A self-aligning joint, using a miniature self-aligning bearing, is shown in *Fig.* 11. This type of joint is used between the adder crank and connecting links. It permits misalignment between the two slides without binding any of the joints. This joint is identical with the radial-bearing type except that only a single bearing is used. The bearing has 5/16-inch O.D., 1/8-inch bore and 7/64-inch width. Because of the self-aligning features, it is not possible to reduce backlash by end-loading this bearing. This can be accomplished only by the preselection of bearings. However, it was found that the backlash was negligible in most of these bearings and preselection was not necessary.

SLIDES: A cross-sectional detail of the slide is shown in *Fig.* 12. The sliding carriage is supported by three balls, hence the name "three-ball" slide. The base of the hardened steel slide is L-shaped and has a ground V-groove in one leg. A closing piece carrying another parallel groove is doweled and clamped to the base. The web portion of the closing piece is narrowed down and cut away in the center leaving two supporting legs. The carriage contains two parallel V-grooves and is supported on three steel balls 1/8-inch in diameter. One ball is located on the closing-piece side of the carriage; the other two on the opposite side. In the clamped position the legs of the closing piece are slightly spring loaded, thereby removing all backlash. A ball retainer, together with a stop, is provided to correctly space the three balls. Depending upon the type of connection made to it, the carriage is either drilled to receive a pin or counterbored to carry a pivot bearing. A slide of this type can be made to track along a straight line within 0.0001-inch. It is used only when the backlash of the ball-bushing type is not acceptable.

ADJUSTABILITY: Shown in *Fig.* 13 is a cavity-tuning mechanism with provisions included for making adjustments. Three parameters are adjustable: link length, crank radius and crank angle. Except for the adjustment features and the slide details, this linkage is similar to the one shown in *Fig.* 1. The link is made in the form of a turnbuckle, and attached to the slide by a pivot-bearing type joint. A radial-bearing type joint is used at the crank end. Adjustment in the length of the link is made by turning the barrel of the turnbuckle, checknuts being provided to lock the barrel. A short slide is provided at the end of the crank, attached to the carriage of which is the pin carrying one end of the link. By moving the carriage in or out, the radius of the crank is varied. This is accomplished by turning a screw, which is threaded into the carriage, against the action of a small compression spring. A clamp is provided to lock the carriage in position. The hub of the crank is attached to a shaft by means of an integral split clamp. Pinned to the shaft above the hub is a collar carrying two adjusting screws. These screws engage a flat portion of the hub. By loosening the clamp and turning one adjustment screw opposite to the other, the crank may

Fig. 11—Below—Detail of self-aligning joint used between the adder crank and connecting links

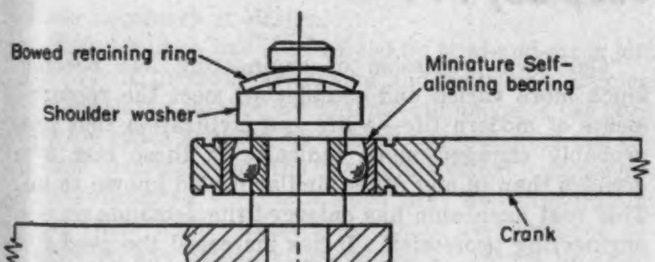
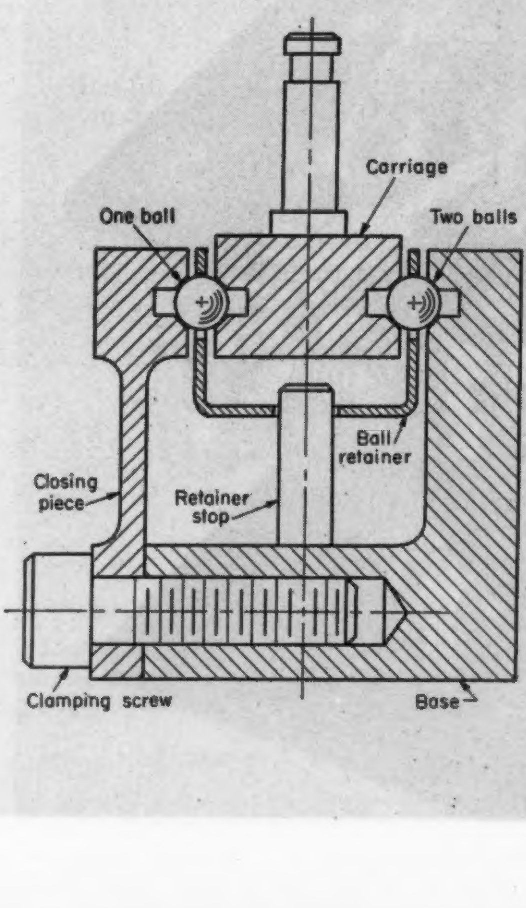


Fig. 12—Right—Detail of three-ball slide used in cavity-tuning mechanism

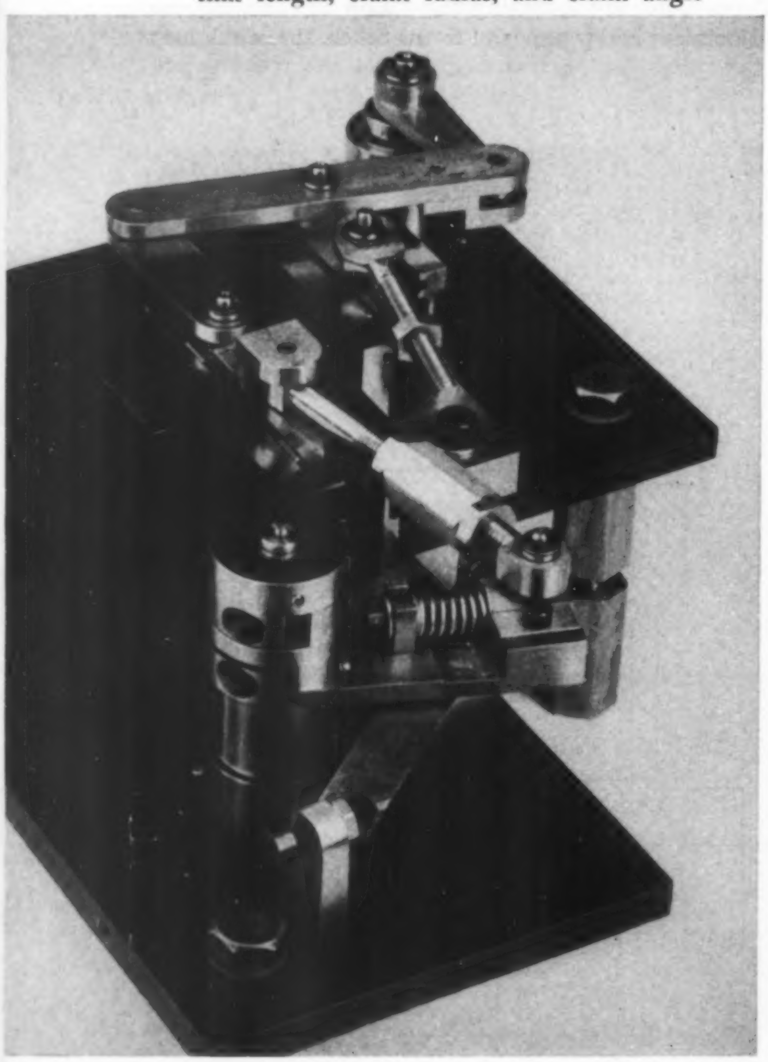


be rotated relative to the shaft.

The mechanism shown is a laboratory model on which measurement of the adjustments is made by optical means. This method would not be suitable on field equipment, and provisions are needed to make these checks by mechanical means. This can be done by providing reference pins and then measuring across the pins. For example, in the link a pin can be located in each end near the joint. By measuring over these pins with a caliper or micrometer, the length can be determined. Similar means can be provided on the crank by locating one pin near the hub and the other on the sliding carriage. Instead of pins, a scale and vernier can be provided on the slide and the change in radius read off directly. Similarly, an angular scale and vernier can be indexed on the adjoining surfaces of the hub and collar to indicate angular adjustment of the crank.

Linkage vs. Cams: Perhaps an obvious approach in designing this tuning device would have been to use a cam-driven mechanism. The shape of the cam would linearize the motion and dither would be introduced by oscillating the cam follower. However, several inherent disadvantages limit its usefulness. Essentially, an infinite number of dimensions determine a cam profile. High accuracy becomes impractical to achieve except at a limited few points. In order to minimize backlash, the follower must be spring-loaded against the cam surface. Then acceleration of the driving members are limited and friction forces become high. The dither mechanism becomes unduly complicated.

Fig. 13—Cavity-tuning mechanism with adjustable linkage, making possible changes in link length, crank radius, and crank angle



Linkage mechanisms do not suffer from these limitations and can be used to far better advantage, considering only functions that are not discontinuous. Linkages are essentially straight members jointed together. Only a small number of dimensions need be held closely. The joints make use of standard bearings and backlash can be minimized by end-loading without inducing high friction loads. High accuracy can be achieved by resorting to jig boring and precision grinding, all done on standard machine tools, and by the preselection of bearings. The links in effect form a solid chain and are not subject to acceleration limitations. The dither requirement is accomplished through a simple eccentric drive. In addition, linkage devices can easily be made adjustable, which greatly increase their usefulness. With cam-driven devices adjustment is highly impractical and very limited. This tuning mechanism gives a practical illustration of the three basic linkage elements: cranks, links and slides. Essentially, any linkage mechanism can be built simply by various combinations and arrangements of these elements.

ACKNOWLEDGMENT: Information in this article is based upon work done under Air Force Contract No. AF-33 (038-13677).

REFERENCES

1. E. W. Pike, T. R. Silverberg and P. T. Nickson—"Linkage Layout," MACHINE DESIGN, Vol. 23, No. 11, November 1951, Page 105.
2. A. Svoboda—*Computing Mechanisms and Linkages*, McGraw-Hill Book Co. Inc., New York, 1948 (Massachusetts Institute of Technology, Radiation Laboratory Series, Volume 27).

They Say . . .

"Our own profession of engineering has become much more varied and complex to meet the requirements of modern life—a life and civilization that has probably changed more radically in these last five decades than in any other similar period known to us. This vast expansion has enlarged the demands on the engineering profession. It has increased the need for articulateness among engineers, for skill and facility in talking and writing—for individual awareness of that comparatively new field called semantics."—LEWIS A. VINCENT, *general manager, National Board of Fire Underwriters.*

"'Research' is a high-hat word that scares a lot of people. It needn't. It is rather simple. Essentially, it is nothing but a state of mind—a friendly, welcoming attitude toward change. Going out to look for a change instead of waiting for it to come."—CHARLES F. KETTERING.

DESIGNING
Right-Angle Helical Gears

... with an exact mathematical method instead of by trial and error or layout

By Oliver Saari
Development Engineer
Illinois Tool Works
Chicago, Ill.

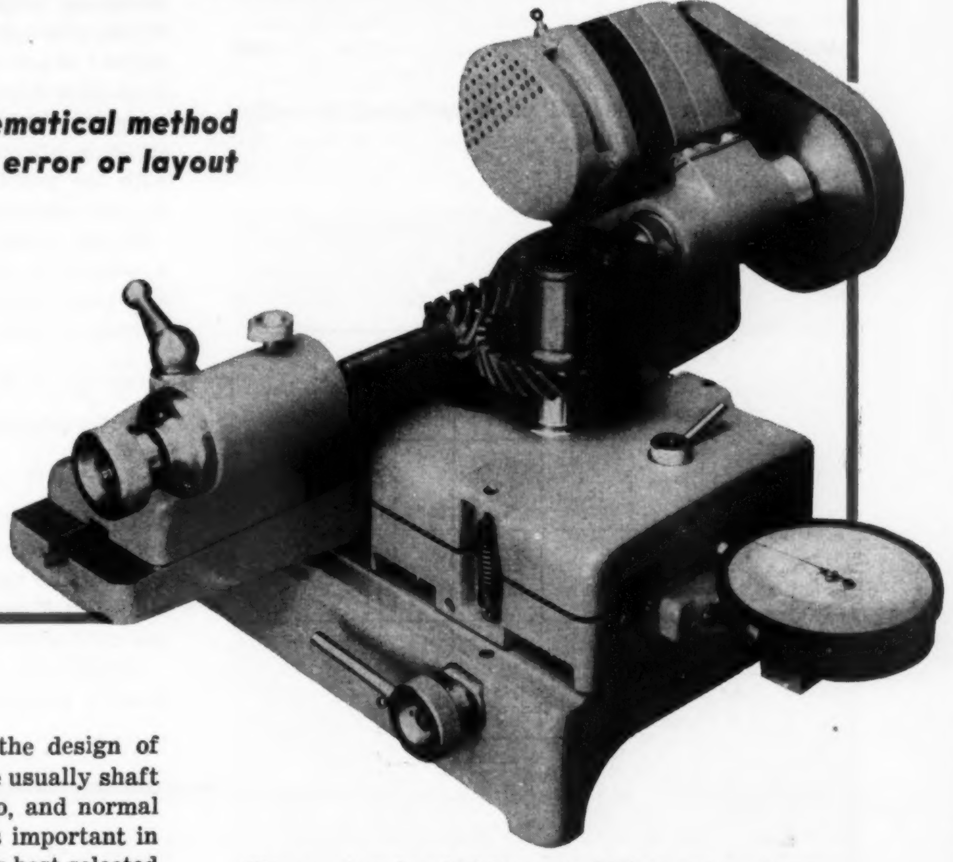


Fig. 1—A pair of right-angle helical gears, shown being checked for runout in a worm-rolling fixture

GIVEN conditions most useful in the design of skew-axis helical gears, *Fig.* 1, are usually shaft angle, center distance, speed ratio, and normal diametral pitch. Position of the axes is important in any design, and normal diametral pitch is best selected to suit available generating tools. Unfortunately, however, this set of conditions leads to an indeterminate equation for the evaluation of remaining conditions necessary in design.

This problem has been solved by trial-and-error and by layout in many textbooks and magazine articles. An ingenious method by Tuplin* circumvents the need for an exact solution by taking advantage of the extreme flexibility of the involute system. However, gears designed by this method are "nonstandard" in that their tooth thicknesses cannot be compared directly on any pair of pitch cylinders. Hence, an exact solution of the problem may sometimes be desirable.

Most skew-axis helical (or "spiral") gears are designed to operate at right angles. Much needless complication is avoided, therefore, if the right-angle case is separated from the general problem, since the formulas are thereby much simplified. This article will show how an exact solution of the right-angle spiral gear problem can be obtained in a straightforward way.

Conventional Relationships: The conventional methods of designing right-angle spiral gears yield the following relationships among the variables shown in *Fig.* 2 and listed in the Nomenclature:

$$R_1 + R_2 = C \qquad (1)$$

$$R_1 = \frac{N_1 \sec \psi_1}{2 P_n} \qquad (2)$$

$$R_2 = \frac{N_2 \sec \psi_2}{2 P_n} \qquad (3)$$

$$\psi_2 = 90 - \psi_1 \qquad (4)$$

$$\sec \psi_2 = \operatorname{cosec} \psi_1 \qquad (5)$$

Combining these equations yields what might be termed the fundamental equation of right-angle helical gearing:

$$\sec \psi_1 + K \operatorname{cosec} \psi_1 = A \qquad (6)$$

where A is a useful symbol for the following relation-

* W. A. Tuplin—"Designing Crossed Helical Gears," MACHINE DESIGN, Aug. 1950, Pages 125-127.

ship among normal diametral pitch, center distance, and number of teeth in the smaller gear:

$$A = \frac{2 P_n C}{N_1} \quad \text{.......... (7)}$$

When the center distance is not important, the helix angle ψ_1 may be given any arbitrary value. Then Equation 6 can be used to find the corresponding value of A, and the center distance can be obtained from Equation 7 rearranged in the form,

$$C = \frac{A N_1}{2 P_n} \quad \text{.......... (7a)}$$

Then the helix angle and pitch radius of the other gear can be found with Equations 3 and 4. This method is essentially the conventional solution given by most sources.

New Approach: When the center distance C is to have a fixed, predetermined value, the problem is not quite so simple, however. Here, Equation 6 must be solved for ψ_1 when A is given, since A is determined by the arbitrary starting conditions. This equation is indeterminate in the sense that when expanded it yields an algebraic equation of the fourth degree, which (though mathematically having an explicit solution) is too cumbersome to use in any practical way. It is this solution which has been obtained by trial-and-error by some and layout by others. Trial-and-error is annoying because one is immediately faced with the problem of where to start. Layout solution by any existing method is of limited accuracy.

It can readily be shown that for Equation 6 to have a solution at all, the value of A must not be less than a certain minimum. The value of ψ_1 which corresponds to this minimum value of A is given by

$$\tan \psi_{1m} = \sqrt[3]{K} \quad \text{.......... (8)}$$

and the minimum value of A itself is given by

$$A_m = \sec^3 \psi_{1m} \quad \text{.......... (9)}$$

Thus both the minimum value of A and the corresponding helix angle ψ_{1m} can be thought of as functions of the speed ratio K. Graphs of these values are shown in ***Figs.*** 3 and 4. The co-ordinates from which the curves are plotted are given in TABLE 1.

Equation 9 permits determination of the maximum number of teeth in the smaller gear as follows:

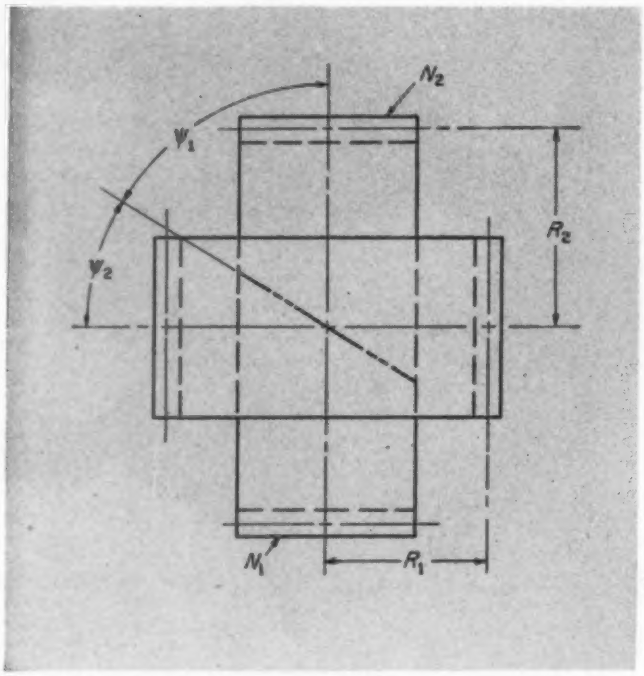


Fig. 2—Above — Relationship of helix angles in right-angle helical gears. The pair shown is proportioned according to the results obtained in the first example

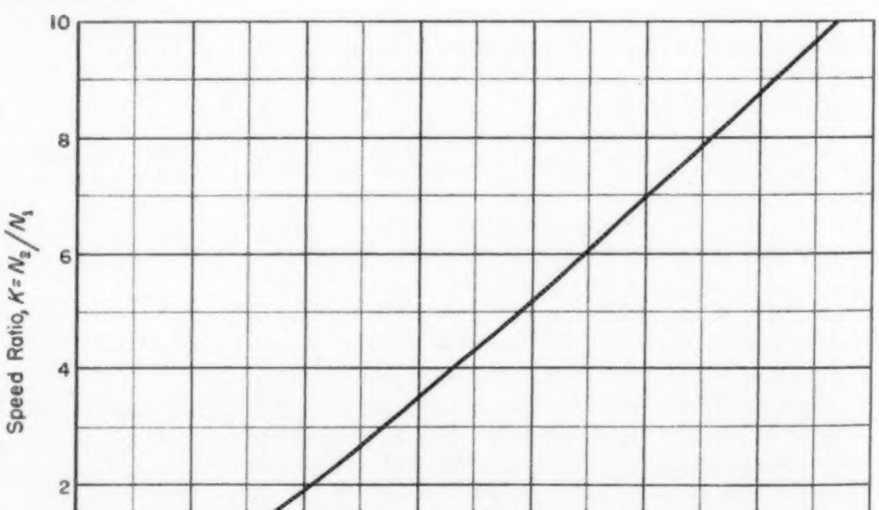


Fig. 3 — Right—Factor A_m versus speed ratio K

$$N_{1\text{-}max} = \frac{2\,P_n\,C}{A_m} \qquad (10)$$

If the smaller gear must have more teeth than this maximum, then the solution is impossible under the given conditions, and either P_n or C must be increased. In general it is best to let the number of teeth in the smaller gear be simply the integer part of the maximum value. This results in gears which make the most effective use of the space allotted them, since they contain the largest number of teeth of the given pitch which can be incorporated into the designated center distance, thereby having higher contact ratios and more tooth-surfaces in action than other designs. However, the design is not limited to this number. Any number N_1 smaller than the maximum may be used.

Having chosen N_1, one must determine the proper value of N_2 to give the desired speed ratio. The obvious relationship is

$$N_2 = K\,N_1 \qquad (11)$$

Choice of N_1 also determines the value of A from Equation 7, and A in turn determines a value of ψ_1 from Equation 6. In fact, there are two values of ψ_1 which satisfy Equation 6 whenever A is greater than the minimum value A_m.

Solutions of Equation 6 can be obtained in a direct way by the use of two formulas.

If ψ_1' is an approximate value of ψ_1,

$$\psi_1' = \psi_{1m} \pm 46.782 \sqrt{\frac{A}{A_m} - 1} \qquad (12)$$

where the units of ψ_{1m} and ψ_1' are degrees.

Equation 12 gives a first approximation which answers the question of where to start. Either plus or minus sign may be used, since there are two solutions. The use of the minus sign results in gears which are more unequal in size. The use of the plus sign gives a design where the diameters of the two gears are more nearly equal, a condition which is perhaps more often desirable. This first approximation can be used directly as the helix angle when the center distance does not need to be exactly the predetermined value.

When greater exactness is desired in the solution, the value of ψ_1' determined with Equation 12 should be substituted into

$$\psi_1 = \psi_1' - 57.296\,\frac{1 + K\cot\psi_1' - A\cos\psi_1'}{A\sin\psi_1' - K\,\mathrm{cosec}^2\,\psi_1'} \qquad (13)$$

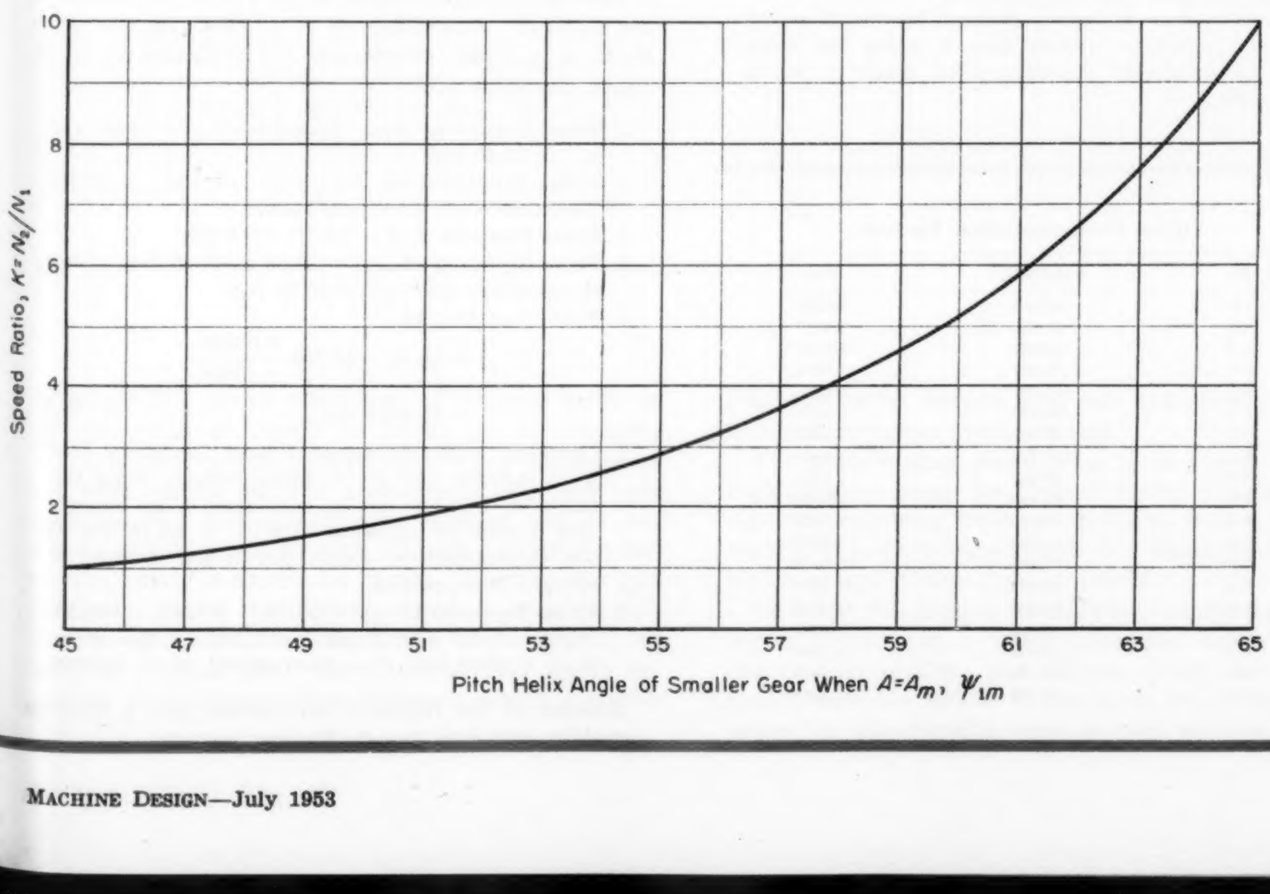
where the angular units again are degrees.

Equation 13 gives a much closer approximation for

Nomenclature

C = Center distance, inches
K = Desired speed ratio = N_2/N_1
N_1 = Number of teeth in small gear
N_2 = Number of teeth in large gear
P_n = Normal diametral pitch
R_1 = Pitch radius of small gear, inches
R_2 = Pitch radius of large gear, inches
ψ_1 = Pitch helix angle of small gear, deg
ψ_2 = Pitch helix angle of large gear, deg

Fig. 4—Helix angles ψ_{1m}, corresponding to A_m in *Fig.* 3, versus speed ratio K



ψ_1, which will be close enough in almost all practical cases, a fact which can be quickly checked in any numerical example by calculating the center distance with Equations 6 and *7a*.

Equation 12 is derived by use of Taylor's expansion about the minimum value of A_m, and Equation 13 is derived from Newton's method of solving indeterminate equations. It is of interest to note that Equation 13 gives a converging series of values of ψ_1, provided each solution is substituted back in as ψ_1'. It might be called a recursion formula in that it alone would give solutions approaching the exact values from any two arbitrary first approximations straddling the ψ_{1m} value. However, the first approximations given by Equation 12 approach the exact values so closely that only one application of Equation 13 is necessary. Thus, the problem is solved explicitly for all practical purposes by means of these two equations.

Design Procedure: When values of K, P_n and C are the given conditions, an outline of the procedure which should be used is as follows:

1. Solve for ψ_{1m} and A_m with Equations 8 and 9, or TABLE 1, or the graphs in *Figs.* 3 and 4.
2. Solve for $N_{1\text{-}max}$ using Equation 10. If the integer part of this number is acceptable as the number of teeth in the smaller gear, use it as N_1. If, however, some other specific number of teeth is desired in the smaller gear (such as a given prime number, or a number containing other prime factors), this number may be used for N_1 provided it is smaller than $N_{1\text{-}max}$. If it is not, then a larger value of P_n or C must be used.
3. Solve for A using Equation 7.
4. Solve for ψ_1' using Equation 12.
5. Solve for ψ_1 using Equation 13.
6. Test this value of ψ_1 by solving for C using Equations 6 and *7a*. If the center distance is within tolerance, as it will be in practically all cases, the solution is complete and the gears can be made with the helix angles determined.
7. If the center distance obtained in step 6 is not within tolerance, repeat step 5, using the value of ψ_1 previously determined in place of ψ_1' in Equation 13.

Table 1—Calculation Factors

K	ψ_{1m} (deg)	A_m
1.0	45.000	2.8284
1.5	48.860	3.5117
2.0	51.561	4.1619
2.5	53.617	4.7812
3.0	55.264	5.4056
3.5	56.630	6.0090
4.0	57.791	6.6037
4.5	58.797	7.1914
5.0	59.681	7.7732
5.5	60.468	8.3501
6.0	61.175	8.9227
6.5	61.816	9.4917
7.0	62.401	10.0573
7.5	62.939	10.6201
8.0	63.435	11.1803
8.5	63.896	11.7382
9.0	64.324	12.2940
9.5	64.725	12.8478
10.0	65.101	13.3998

As previously mentioned, these equations are for the right-angle spiral gear problem only. The method can be extended to cover cases where the axes are at some angle other than 90 degrees. However, since the resulting equations contain the shaft angle as a variable, they are more complicated.

Examples: Several numerical examples might be solved to show how the equations apply. In the first one let the given conditions be $C = 4.000$ inches, $K = 2$, and $P_n = 8$. The steps in the calculation to find ψ_1 are:

1. From TABLE 1, $A_m = 4.1619$ and $\psi_{1m} = 51.561$ deg.
2. From Equation 10, $N_{1\text{-}max} = 2(8)(4)/4.1619 = 15.38$. $N_1 = 15$ can be used; therefore, $N_2 = 30$.
3. From Equation 7, $A = 2(8)(4)/15 = 4.26667$.
4. From Equation 12,

$$\psi_1' = 51.561 \pm 46.782\sqrt{\frac{4.26667}{4.1619} - 1}$$

$$= 51.561 \pm 7.422 \text{ deg}$$

With the plus sign, $\psi_1' = 58.983$ deg.

5. Let $\psi_1' = 58.98$. Since this value is only the first approximation, it may be rounded to the nearest angle whose functions can be found without interpolation in trig tables. No loss of accuracy results. From Equation 13,

$$\psi_1 = 58.98 - 57.296\frac{1 + 1.20267 - 2.19877}{3.65648 - 2.73210}$$

$$= 58.98 - 0.2417 = 58.7383 \text{ deg}$$

6. Checking with Equations 6 and *7a*, $A = 1.926959 + 2(1.169860) = 4.26668$; $C = 4.26668\ (15)/16 = 4.00001$.

This accuracy far exceeds practical requirements. Usually fewer decimal places can be employed in the calculations; more were used here merely to demonstrate the power of the method.

Another example shows some minor variations in the method. This time, let $C = 4.000$, $P_n = 8$, $K = 30/13 = 2.30769$. Therefore, $N_1 = 13$ and $N_2 = 30$. Again the steps are:

1. From curves in *Figs.* 3 and 4, $A_m = 4.55$ and $\psi_{1m} = 52.88$ deg.
2. From Equation 10, $N_{1\text{-}max} = 64/4.55 = 14.07$. Therefore $N_1 = 13$ is permissible.
3. From Equation 7, $A = 64/13 = 4.92308$.
4. From Equation 12, $\psi_1' = 52.88 \pm 13.40$ deg. Using the plus sign gives $\psi_1' = 66.28$ deg.
5. From Equation 13,

$$\psi_1 = 66.28 - 57.296\frac{0.03358}{1.75398}$$

$$= 65.1831 \text{ deg}$$

6. Checking with Equations 6 and *7a* shows that $A = 4.92502$ and $C = 4.0016$ inches. Since this value of C might not be considered close enough, step 5 shall be repeated with $\psi_1' = 65.18$ deg as the approximation. Again places are dropped to obviate interpolation.
7. From Equation 13, $\psi_1 = 65.18 - 0.0249 = 65.1551$ deg. A check with Equations 6 and *7a* now shows $A = 4.92308$ and $C = 13(4.92308)/16 = 4.00000$.

Results of the repeated calculation give a solution certainly accurate enough for any purpose.

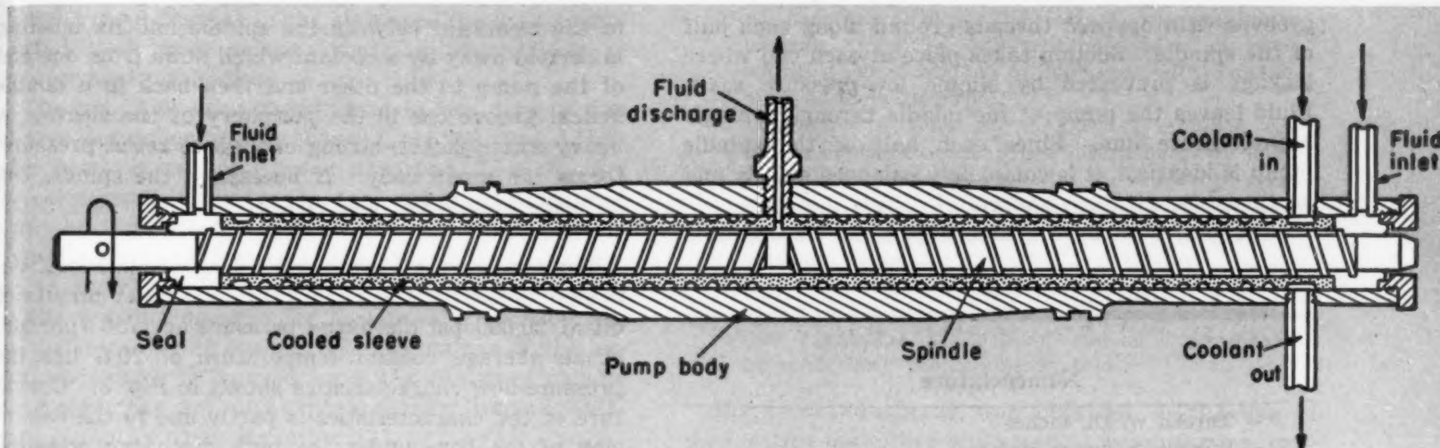


Fig. 1 — Above — Section through spindle drag pump. Coolant passage is a helical groove in the outer surface of the sleeve

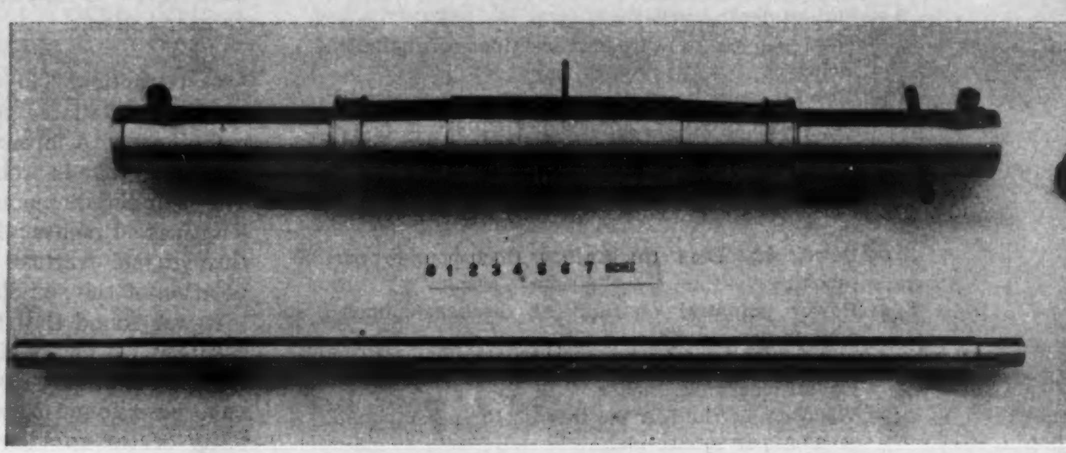


Fig. 2 — Right — Spindle drag pump with spindle removed, showing simplicity of the design. Extremely shallow threads on spindle are nearly invisible

Spindle Drag Pump

. . . employs extruder principle for small discharge of viscous fluids at high pressure

By R. A. Strub*
Senior Research Associate
Polychemicals Dept.
E. I. du Pont de Nemours & Co. Inc.
Charleston, W. Va.

PUMPING small quantities of viscous fluids at pressures of 10,000 to 30,000 psi is ordinarily done with piston pumps which, when designed for use with viscous fluids at pressures in this range, are complicated. Mechanical parts such as pistons and valves are subjected to high alternating stresses and require costly maintenance work. Further, the pulsating nature of the discharge may be objectionable because of possible fatigue failure of piping and the need for regular flow. The latter is often highly important in certain chemical processes where it is necessary to inject viscous fluid into the process stream of a high pressure synthesis unit.

The spindle drag pump, *Fig.* 1, is a new type of high-pressure pump for viscous fluids. Simple in design and requiring little maintenance, it is a rational approach toward overcoming the disadvantages of piston pumps in this type of service.

Basically the pump is an extruder. A spindle, rotating in a sleeve at a speed which may reach several thousand revolutions per minute, drags the viscous fluid toward the middle of the pump by the combined action of the rotation and of two shallow helical

*Now associated with Sulzer Brothers, Winterthur, Switzerland.

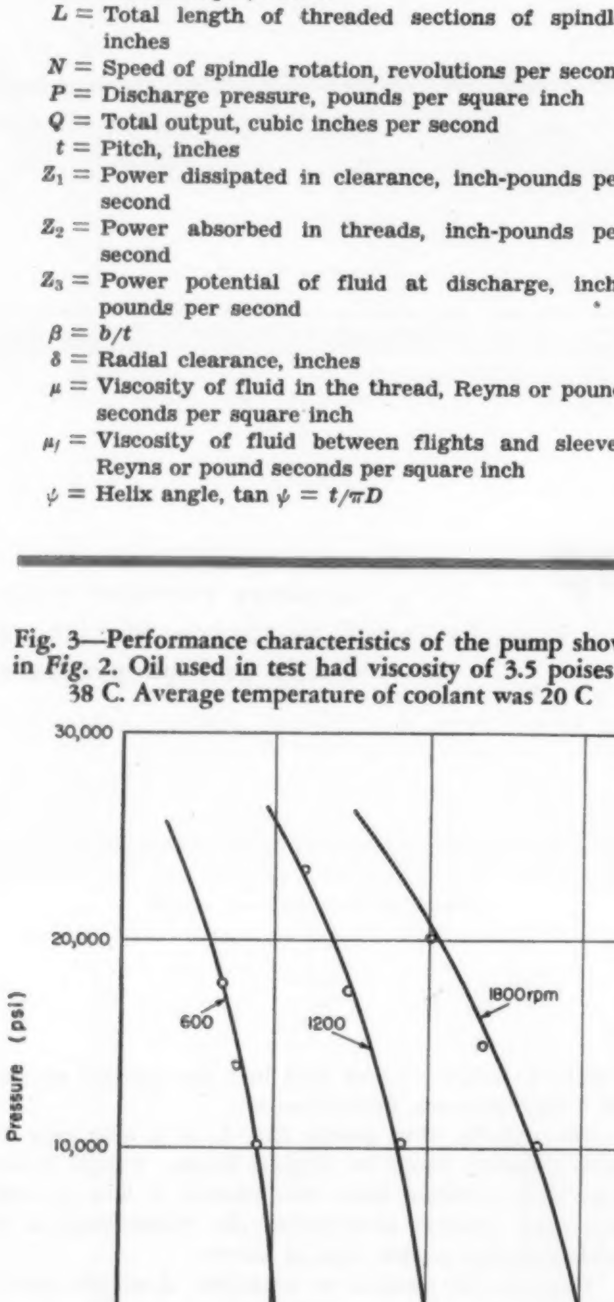
grooves with opposed threads ground along each half of the spindle. Suction takes place at each end where leakage is prevented by simple low-pressure seals. Fluid leaves the pump at the middle through a single high-pressure line. Since each half of the spindle pump is identical, it is completely balanced axially and no thrust bearing is required.

Heat generated by the shearing of the viscous fluid in the clearance between the spindle and its bushing is carried away by a coolant which flows from one end of the pump to the other and then back in a double helical groove cut in the periphery of the sleeve. A heavy water jacket, strong enough to resist pressure, forms the pump body. If necessary, the spindle can be made hollow and also cooled.

Nomenclature

b = Thread width, inches
D = Spindle diameter, inches
e = Flight width, inches
h = Thread depth, inches
L = Total length of threaded sections of spindle, inches
N = Speed of spindle rotation, revolutions per second
P = Discharge pressure, pounds per square inch
Q = Total output, cubic inches per second
t = Pitch, inches
Z_1 = Power dissipated in clearance, inch-pounds per second
Z_2 = Power absorbed in threads, inch-pounds per second
Z_3 = Power potential of fluid at discharge, inch-pounds per second
$\beta = b/t$
δ = Radial clearance, inches
μ = Viscosity of fluid in the thread, Reyns or pound seconds per square inch
μ_f = Viscosity of fluid between flights and sleeve, Reyns or pound seconds per square inch
ψ = Helix angle, $\tan \psi = t/\pi D$

Fig. 3—Performance characteristics of the pump shown in *Fig.* 2. Oil used in test had viscosity of 3.5 poises at 38 C. Average temperature of coolant was 20 C



Performance: A large spindle drag pump, *Fig.* 2, designed to deliver 200 cubic centimeters per minute of oil at 20,000 psi discharge pressure at 1750 rpm and at an average coolant temperature of 20 C has the pressure-flow characteristics shown in *Fig.* 3. Curvature of the characteristics is partly due to the expansion of the bore under the high discharge pressure. This leads to greater clearances and therefore to greater leakage.

Construction: The pump is built entirely from commercially available tubing, thus reducing the cost of machining. A bronze sleeve with a double helical cooling groove is a shrink fit in the heavy pump body. The spindle diameter is 1⅜ inches and its overall length is 42 inches. It is driven by an elastic coupling through an overload shear pin which fixes the axial position of the spindle with respect to the pump body.

It was found that long holes could be machined accurately by using standard boring bars and honing tools. Small bows of a few thousandths of an inch have been found acceptable because of the high flexibility of the spindle.

The pump has the inherent advantages of being simple and requiring low maintenance. A similar pump operated for several thousand hours before it was necessary to replace the spindle. Wear of the spindle or the bushings can be compensated for by electrolytic plating of the spindle.

Design: Spindle drag pump design is based on the theory of screw extruders[1, 2]. For a uniform shallow thread profile the pressure is related to the output by the equation (see Nomenclature)

$$P = \left(\frac{3 \pi D N}{h^2 \tan \psi} - \frac{3 Q}{\pi \beta h^3 D \sin^2 \psi} \right) L \mu \qquad (1)$$

The output Q is maximum when p is made equal to zero. The maximum pressure occurs at zero output.

Leakage past flights diminishes the pressure generated by the pump.[2] From test results the average viscosity including the influence of leakage can be calculated by using Equation 1.

The power absorbed is given by the relations[2]

$$Z_1 = \pi^3 D^3 N^2 \frac{L}{t} \frac{e}{\delta} \mu_f \qquad (2)$$

$$Z_2 = \pi^3 D^3 N^2 \frac{L}{t} \frac{b}{h} \mu \left(4 - \frac{3 Q}{Q_{max}} \right) \qquad (3)$$

In Equations 2 and 3 two different values for the viscosity have been introduced. Viscosity μ_f refers to the fluid situated between the flights and the sleeve. This viscosity is generally lower than the viscosity μ

1. References are tabulated at end of article.

in the thread because in the clearance the fluid is submitted to a more intense shearing which leads to greater heat dissipation.

The potential energy contained in the fluid at discharge corresponds to a power $Z_3 = PQ$. The difference between the total power absorbed, $Z_1 + Z_2$, and Z_3 corresponds therefore to the energy to be carried away by the coolant.

In the previous equations the viscosity to be introduced is difficult to predict. It is generally a function of the working pressure, of the speed of rotation and of the intensity of cooling. Proper values of the average viscosity to be used for a new design of spindle drag pump can be found experimentally from a laboratory pump by using Equation 1.

Theoretical work has been carried out in an attempt to correlate the helix angle, ψ, to the thread depth, h, for optimum pump design. Although several combinations of speed, helix angle and thread depth are possible, it has been found that the following equation is a good guide for relating the different variables:

$$\tan \psi = \frac{2\,Q}{\beta\,\pi^2\,D^2\,Nh} \qquad (4)$$

The length of the pump is then determined by Equation 1.

Test Results: An extensive test program has been carried out to find the influence of the different factors affecting the pump design. A spindle 11 inches long and ¾ inch in diameter was spun at speeds varying from 1000 to 5000 rpm in a water-cooled bronze sleeve. Thread depths from 1 to 6 thousandths of an inch and pitches from ½ to 1⅜ inches (helix angles from 12 to 30 degrees) were tested. The radial clearance was of the order of ½ thousandth of an inch.

Test results with different oils are summarized below. The conclusions have been found to be also generally valid for a larger spindle pump 1⅜ inches in diameter by 34 inches long.

1. The output at no pressure varies proportionally with speed and thread cross section.
2. At constant flow of the coolant, the maximum pressure at no flow is nearly independent of speed within a range of 1000 to 3000 rpm with a maximum at about 2000 rpm.
3. Maximum pressure at no flow is obtained for thread depths from 2 to 4 thousandths of an inch.
4. Power absorbed increases with the discharge pressure and speed. For temperatures of the coolant from 10 to 25 C, the power absorbed is nearly proportional to speed.
5. If the cooling intensity is increased, the pressure or the output is raised markedly. This indicates that flow regulation of the pump can be accomplished by controlling the cooling system.

No great advantage was obtained in the use of double threads either with respect to output or pressure generated, although in some other applications heat transfer is increased by a larger number of threads in parallel. Further, for optimum design, it is best to use a constant pitch along the spindle.[2] Flight width has been found to be noncritical but must be large enough to provide sufficient bearing surface.

Proposed Design for Higher Pressures: A proposed design for pressures in excess of 30,000 psi is shown in *Fig.* 4. Pressures of this magnitude would be valuable for testing plant equipment or autofrettaging heavy cylinders. Any desired pressure can be maintained by adjusting the intensity of cooling.

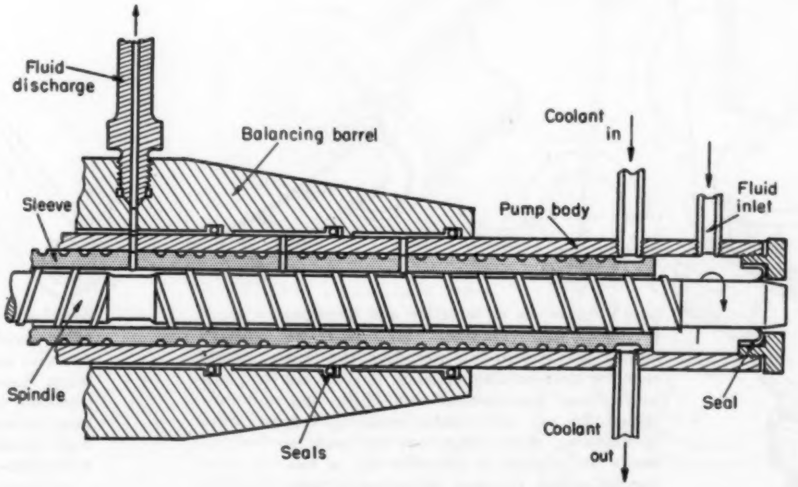
To compensate for excessive dilation of the bore at high pressure, a series of balancing chambers are distributed along the balancing barrel. Each of these chambers, sealed by rubber rings and anti-extrusion washers, are pressurized by the fluid being pumped. Discharge pressures of the order of 100,000 psi could be reached with such a pump.

Acknowledgment is due Mr. P. R. Weber, engineering department, for his helpful suggestions and for promoting the development of the pump.

REFERENCES

1. H. S. Rowell and D. Finlayson—"Screw Viscosity Pumps," *Engineering*, August, 1928, Pages 249, 250 and September 1928, Pages 606, 607.
2. R. A. Strub—"The Theory of Screw Extruders," *Ohio State University Bulletin No.* 149, 1952, Pages 481-494.

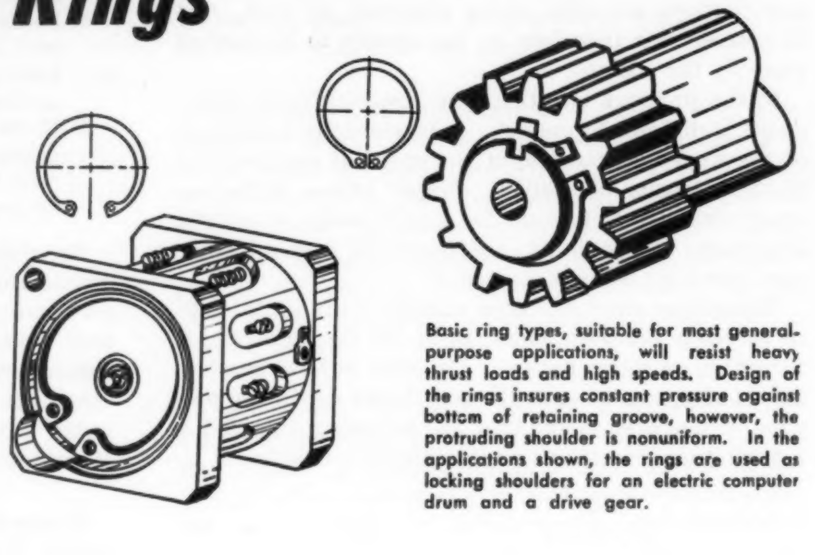
Fig. 4 — Pump design for discharge pressures to 100,000 psi. Fluid between balancing barrel and sleeve prevents excessive bore expansion at high pressures



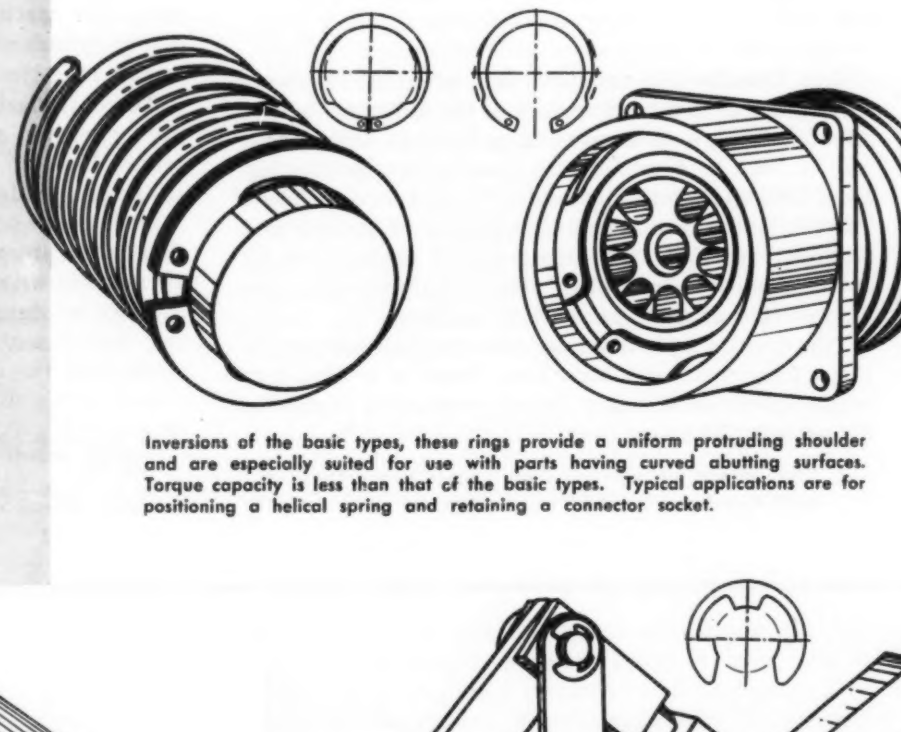
DESIGN DETAILS

Retaining Rings

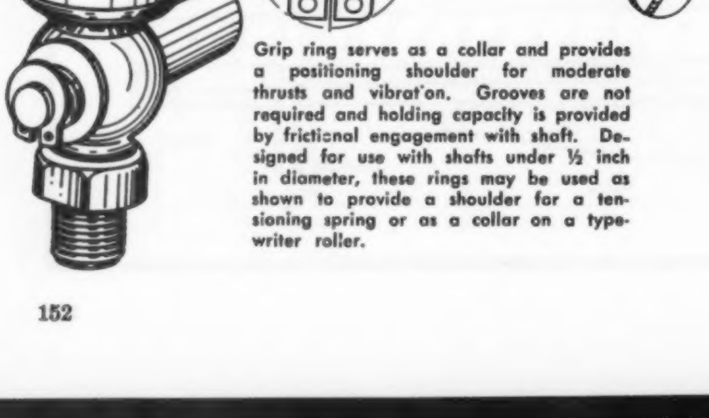
RETAINING rings offer a simple solution to many of the fastening and positioning problems commonly encountered in design. Snapped into a circular groove on a shaft or in a housing, they provide a resilient artificial shoulder for accurate locating, retaining or locking of component parts in machine assemblies. A relatively large degree of spreading can be absorbed by the rings without deformation providing compensation for tolerance variations in groove dimensions. Some typical and unusual applications of standard types of Truarc retaining rings are shown accompanying. A product of Waldes Kohinoor Inc., these rings afford a range of selection in meeting diverse application requirements such as high speeds, impact and vibration loads, shaft inaccessibility, end play takeup and ease of maintenance. In addition, self locking types available can often be utilized without grooves for applications involving moderate loads. Many special and modified types are available.



Basic ring types, suitable for most general-purpose applications, will resist heavy thrust loads and high speeds. Design of the rings insures constant pressure against bottom of retaining groove, however, the protruding shoulder is nonuniform. In the applications shown, the rings are used as locking shoulders for an electric computer drum and a drive gear.



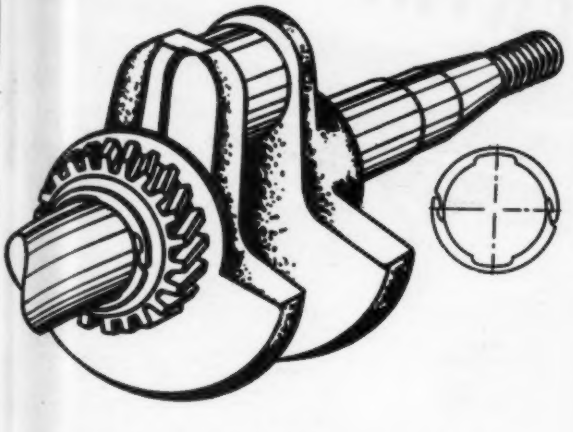
Inversions of the basic types, these rings provide a uniform protruding shoulder and are especially suited for use with parts having curved abutting surfaces. Torque capacity is less than that of the basic types. Typical applications are for positioning a helical spring and retaining a connector socket.



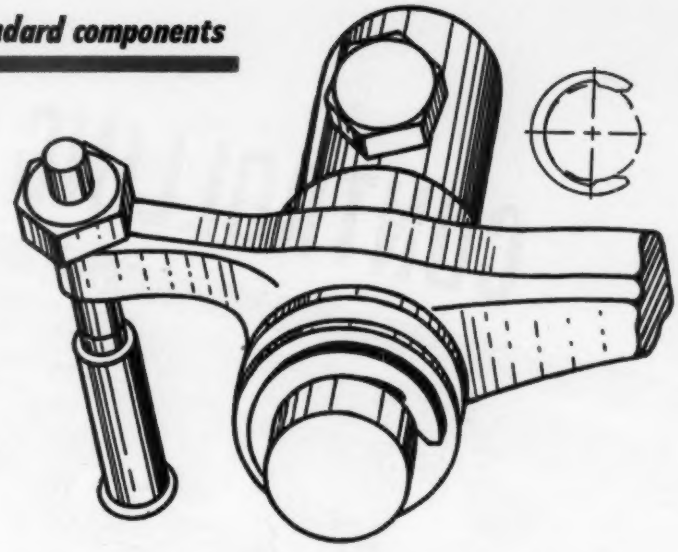
Grip ring serves as a collar and provides a positioning shoulder for moderate thrusts and vibration. Grooves are not required and holding capacity is provided by frictional engagement with shaft. Designed for use with shafts under ½ inch in diameter, these rings may be used as shown to provide a shoulder for a tensioning spring or as a collar on a typewriter roller.

Attached radially, these open type "e" rings provide a large shoulder for shafts under 1½ inches in diameter and are also applied in bowed construction for end play takeup. Rings have been designed to withstand spreading during assembly and disassembly and can be used with a relatively deep groove. These rings are also not recommended for high speed operation. Typical of the possible applications is a heavy-duty pantograph linkage.

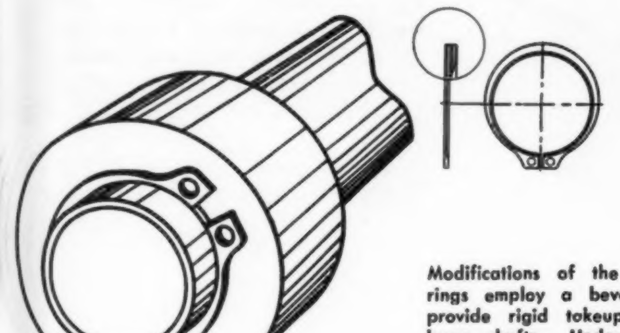
Practical solutions for design problems with standard components

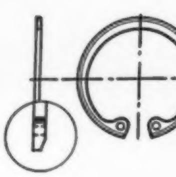


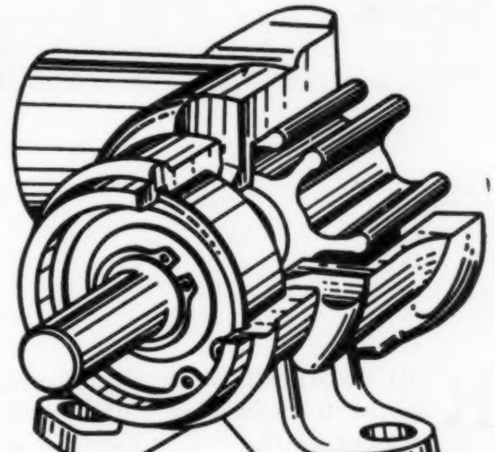
Interlocking ring for axially inaccessible shafts will withstand heavy thrust load and extremely high speeds. Installed in a deep groove, this ring provides a uniform protruding shoulder. Two identical semicircular halves are attached radially and interlock at the center. Application shows use of ring as a locking shoulder for a drive gear.



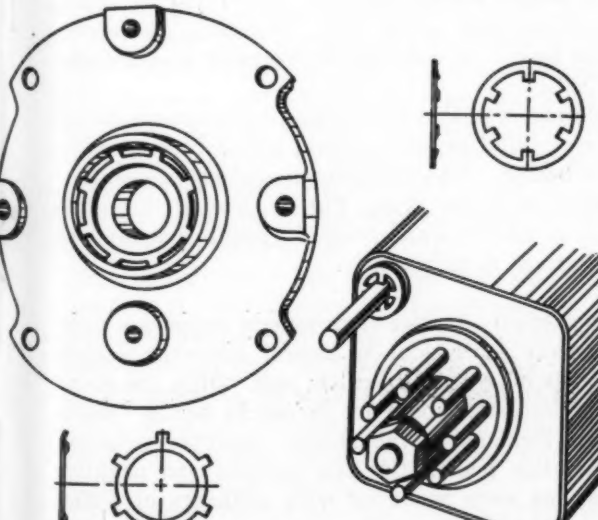
Crescent ring is especially suited for axially inaccessible shafts. Radially applied, this ring provides a uniform shoulder and can be used with a deep groove for protection against impact and vibration loads. Also available in bowed construction for end play takeup. Loosening of the ring caused by excessive deformation can be dangerous; these rings are not recommended for high speed operation. Illustrated is the application of the ring as a shoulder on a rocker arm.



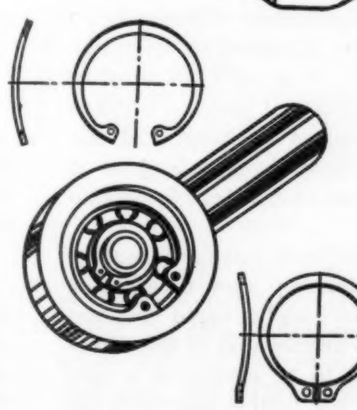


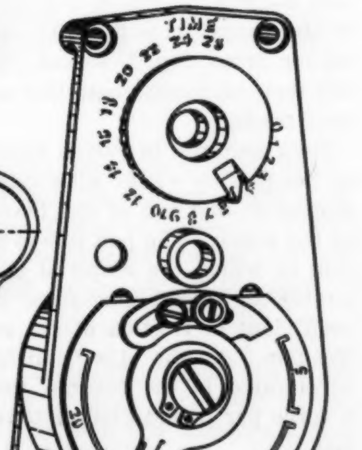


Modifications of the basic types, these rings employ a beveled construction to provide rigid takeup of end play for large shafts. Under load the beveled edge acts as a wedge to lock the abutting part. Illustrations show application of rings to a preloaded assembly and a pump housing.



Self-locking rings require no grooves and are recommended for permanent assemblies where thrust, impact and vibration loads are moderate. Inclination of prongs provides holding capacity in one direction only. This holding capacity can be tripled by using a shallow groove for mounting. Applications show use as positioning shoulders for a rubber bushing and for plastic pins.





Modifications of the basic types, these bowed rings for shafts and housing diameters under 1½ inches provide resilient takeup of end play as well as damping of vibrations and oscillations. Groove width must be sufficient to accommodate bow. Possible applications are as a shoulder for a preloaded bearing and to provide variable tension on the indicator of an altimeter.

CONTROLLING INERTIA

By W. W. Sloane
Vice Pres. in Charge of Eng.
Goodman Mfg. Co.
Chicago, Ill.

Fig. 1—Shaker conveyor unit which reciprocates a trough line up to 300 feet long for moving bulk material

REQUIREMENTS of shaker conveyors have led to the development of a drive incorporating irregular gearing for producing a specific acceleration pattern in reciprocating motion. Consisting of a long reciprocating trough line mounted on rollers or ball bearings, a shaker conveyor is used for moving bulk material in mines and industrial plants. Strokes in the range from 6 to 12 inches are used and one shaker drive, *Fig.* 1, serves trough lines in lengths to 300 feet, although installations to 600 feet have also been made.

Bulk material is moved with back and forth motion of the pan by virtue of a relatively slow reversal of motion at one end of the travel and a rapid reversal at the other. The pan line is accelerated in the direction in which the material is to be moved at a rate having an accelerating force per pound less than the coefficient of friction of the material in contact with the pan. At the other end of the stroke the pan is accelerated in the reverse direction at a rate giving a force per pound greater than the coefficient of friction.

The ratio of the accelerating force in one direction to that in the other—the kick ratio—is a measure of the conveying rate. Kick ratios as low as 1.5/1 and as high as 4.5/1 have been designed. Although a high kick ratio enables the drive to move material up steeper grades and faster on the level, it also imposes higher stresses per foot of pan line. Generally, therefore, permitted length of pan line is inversely proportional to kick ratio.

The character of this conveying process immediately suggests certain dynamic criteria which become paramount in design. How ideal kinematic objectives have been achieved in the drive, *Fig.* 2, through the combined use of linkages and irregular profile gearing will be summarized in this article.

System Origin: Service environment imposes a number of severe limitations in shaker conveyor design. One limit is height of the entire unit which, for many mine installations, should not exceed 24 inches. Early European versions of the shaker conveyor principle were basically pneumatic; the positive and negative accelerations were produced with cylinders of different diameter from a common pressure. However, because of the required auxiliary equipment, combined again with service conditions, attention in this country was directed toward a compact, self-sufficient mechanical drive.

A reasonably satisfactory solution, compatible with size and service requirements, was found in linkages.

How linkages and irregular gearing are combined to achieve improved performance in shaker conveyors

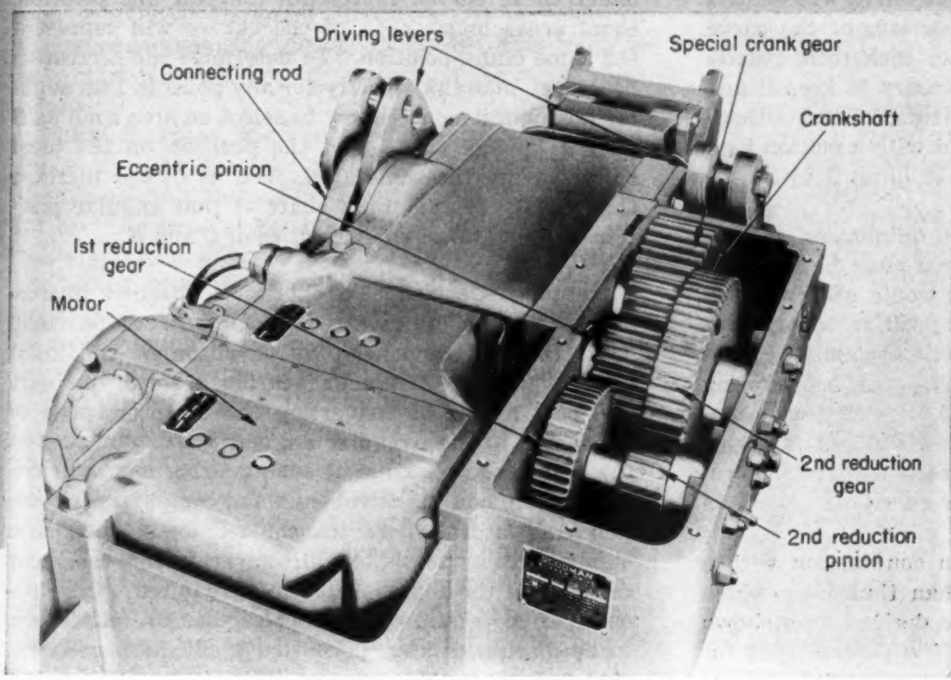


Fig. 2—Drive assembly incorporating speed reduction gears and linkage. Last reduction pair is irregular in profile, providing a variable input to the linkage for achieving most favorable inertia properties

By means of a somewhat complex linkage system, the velocity characteristic shown in ***Fig.*** 3 was obtained. ***Fig.*** 4 shows the corresponding acceleration curve.

The velocity curve is helpful in the visualization of three features: position of pan line, velocity, and acceleration. The area under the velocity curve is the measure of the position of the pan line, the height of the curve the velocity, of course, and the slope of the curve, as measured by the tangent of the angle to its base, the acceleration.

Velocity of the sliding material can also be represented on the velocity curve. Shown on ***Fig.*** 3, it is drawn toward the right from the peak of the curve to its intersection with the lower part of the curve. Its slope (tangent of the angle) is equal to the coefficient of friction of the material, at the same scale that the force of acceleration is measured (slope or first derivative of the velocity curve).

The ordinate of the line is a measure of the velocity of the material relative to the ground. Besides this absolute velocity, the velocity of the material relative to the pan is represented by the ordinate distance between the straight line and the curve. Also the area between the line and the curve represents the distance which the material advances in one cycle.

Moving the largest amount of material with the lowest stresses in the pan line and the drive requires that four factors receive attention in design:

1. The distance which the material can be moved per cycle.
2. The number of cycles per minute at which the drive can be operated without having the material backslide on the slow accelerating stroke.
3. The maximum forces developed by the rapid reversal of the pan line.
4. The shock to the pan line and mechanism by the reversal of forces.

Analysis of ***Fig.*** 3 with these factors in mind reveals some pertinent facts. The larger angle α is made, the greater becomes the area between the material velocity line and the shaker velocity curve. But also, the larger angle α, the higher the inertia stresses in the pan line.

Controlled Inertia: It becomes obvious that the most effective combination of these diverging objectives—high material velocity and low acceleration—can be best realized if the velocity curve is designed with straight-line portions joining the peaks. In the velocity curve of ***Fig.*** 3 this ideal is not satisfied; the initial portion of the curve shows departures from a straight line. Because of these fluctuations, the speed

of the drive must be held down. Otherwise, the higher accelerations occurring at some points would cause backsliding of the material.

The straight-line velocity sections are, of course, uniform acceleration stages. Hence, acceleration would be the absolute minimum for the prescribed conditions, and inertia forces would correspondingly be limited to their lowest possible values.

Making the center portion of the curve a straight line moves the peaks toward the center and a straight line on the left-hand portion of the curve will be at a smaller angle than the steeper portions of the curve as shown. Consequently a higher kick ratio results and further adjustments are necessary to keep it and the stresses within preselected limits. These adjustments can be made by starting out with a motion having a lower kick ratio and then building it up to the desired value.

Although uniform acceleration minimizes the inertia loads, some further measures must be taken to eliminate the shock loading that would otherwise occur in the instant change from positive to negative acceleration. In such a high inertia system it is mandatory that "jerk"—the rate of change of acceleration —be maintained at a finite value. Specifying a constant finite value for jerk produces a sloped straight-line acceleration region and a second-degree (parabolic) transition in the velocity curve, as shown in *Fig.* 5.

This concept was developed in conjunction with a much simpler linkage, *Fig.* 6, than that from which the curves of *Figs.* 3 and 4 were derived. As shown by the linkage velocity curve in *Fig.* 5, developed for a low kick ratio application, the simplified linkage yielded characteristics that are a reasonable first approximation of the ideal velocity curve based on the principles just discussed. The slope of the ideal velocity curve is, of course, the trapezoidal acceleration diagram.

The linkage curve is based on the uniform velocity of the crankshaft. The base of these curves is time but so far as the linkage curve is concerned it could also be crank position. The base does not represent crank position for the ideal curve. The difference curve shows the change in the pan line velocity necessary to produce the ideal curve. It is plotted to time and does not apply to crank position. Since the area under these curves represents the pan line position, equal areas under each of the curves will represent the same crank position. To determine the correction necessary in crank velocity for any point in this angular position, it is necessary to select an area such as *B* on the linkage curve, find the position on the ideal curve having the same area, such as *C*, and increase the velocity of the crank shaft at that angular position in the ratio of the velocities of *C* and *B*.

Irregular Gearing: To obtain the desired linkage output motion, the required velocity correction could be introduced, obviously, by variation of the input crank velocity. Irregular gearing seemed the only and best means of converting the uniform speed of the motor to the irregular angular velocity needed at the crank. Application of such gearing has been relatively common in instruments and computers for motion translators where transmitted loads have been either small or negligible. However, in this case gear capacity in the region of 60 hp is required. Development of successful design and production techniques, to be discussed later, permitted application of irregular gearing to the conveyor drive.

Space limits added another consideration at this point. Irregular gears having an average 1/1 velocity ratio could have been used. But to reduce the num-

Fig. 3—Velocity diagram typical of earlier complex shaker linkages driven by constant-velocity gears

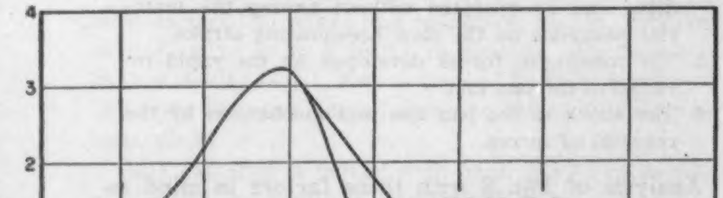
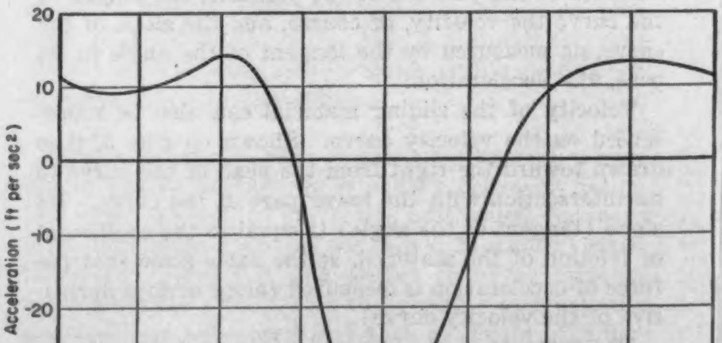


Fig. 4—Acceleration diagram corresponding to the velocity pattern of *Fig.* 3



ber and size of gears required in the speed-reducing drive, obtaining some speed reduction, besides modifying the constant motion, became another objective in the design of the irregular gears. An average 2/1 reduction was found to be feasible and proved quite satisfactory in the drive, *Fig.* 7.

In this arrangement the pinion makes two revolutions while the crankshaft makes one. The speed corrections necessary are quite different during the two pinion revolutions, *Fig.* 8. At any point on the first revolution curve, the same pinion tooth is in engagement as at the point directly above or below on the second revolution curve. Of course, this tooth is in contact with different gear teeth on the two revolutions.

The differences in speed are obtained by design of the gear teeth to operate at different pitch radii on the pinion, on which all teeth are of identical standard form. The general swings of the curves above and below the base line, *Fig.* 8, are obtained by mounting the pinion eccentrically. Thus a standard pinion is used, with its theoretical or nominal pitch circle eccentrically rotated, but moreover its true pitch line of engagement deviates from the circle according to the special pitch profile of the mating gear. Correspondingly, the true pressure angle varies between certain narrow limits.

The required shapes of the gear teeth are developed for adequate tooth contact and strength. Every tooth in the gear is different and no tooth has the same shape on both sides. The gear is produced in a special fixture mounted on the spindle of a Fellows gear shaper. A secondary spindle carrying the gear blank,

Fig. 5—Plots of relationships involved in the determination of optimum inertia characteristics

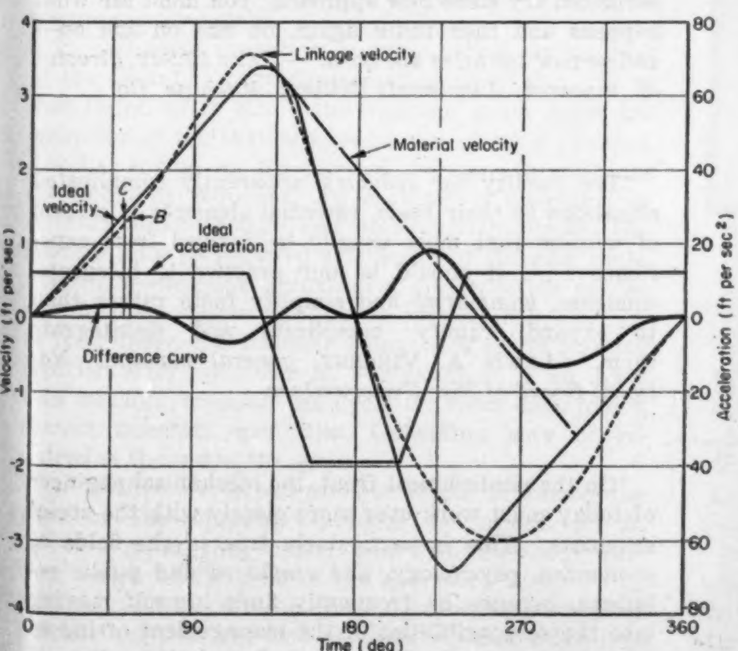


Fig. 6—Simplified linkage system adopted in conjunction with the irregular gear drive

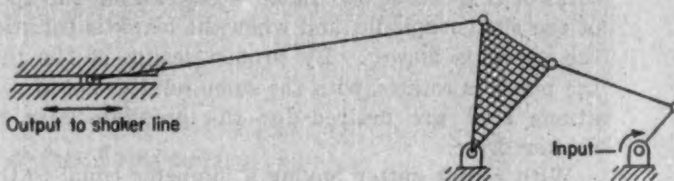


Fig. 7—Irregular gear pair having an average 2/1 ratio. Pinion is circular spur with standard tooth form, but eccentrically mounted. Gear teeth are all different and provide speed adjustment at the input crank of the linkage, Fig. 6

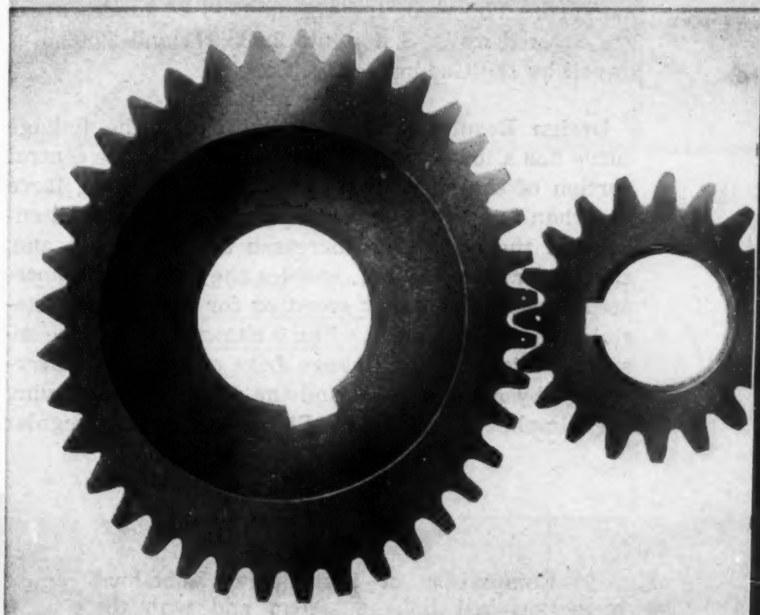
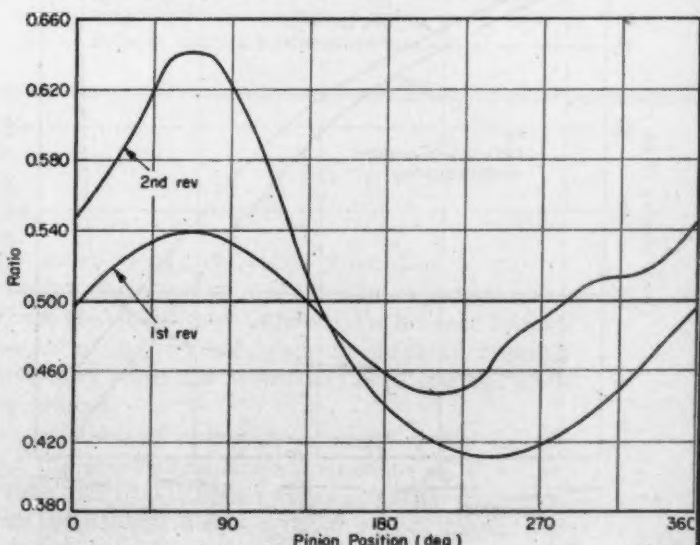


Fig. 8—Curves showing the ratio adjustment provided by the irregular gears for the two revolutions of the pinion occurring during one revolution of the gear



is so arranged that it can be advanced or retarded rotationally by means of a stationary cam. As the shaper spindle revolves and the blank advances, the rotational speed of the blank is faster than the speed of the shaper spindle, and when the blank is retarded, the speed is slower. By proper design of the cam, the blank is rotated with the same relative speed variations that are desired for the crankshaft in the shaker drive.

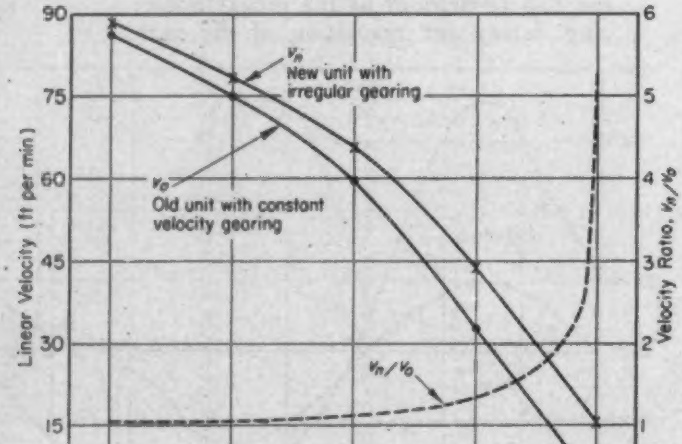
With a disk cutter having a diameter equal to that at the working depth of the pinion and mounted eccentrically, the desired irregular contour of the gear blank is obtained. After the blank is formed, the disk cutter is replaced with a toothed cutter having the same tooth outline and number of teeth as the pinion. The cutter and pinion teeth are of involute form, not modified. The gear and the method of producing it are covered by U. S. Patents 2,585,971 and 2,602,374, owned by the Goodman Mfg. Co.

Design Results: As shown in ***Fig.*** 5 the linkage curve has a low kick ratio and the slope of the central portion of the curve represents an accelerating force less than is acceptable for a final design. Straightening out the curve has increased the kick ratio and, as the curve is laid out, enables the drive to be operated at either a higher speed or for conveying material up a steeper grade. ***Fig.*** 9 shows comparative results obtained with a linkage drive of the type characterized by ***Figs.*** 3 and 4 and the new drive employing the simplified linkage of ***Fig.*** 6 with the irregular gears. This figure shows results when both drives are running at the same speed. The new drive can be run at a higher speed than the old ones without back slipping. When operated at a higher speed for transmission of material on the level, the stresses are increased, but the material movement is much faster.

When a drive is operated at its maximum efficiency it must be operated at a slower speed when the material is moved up hill and the force of acceleration must be kept below the coefficient of friction after the percentage of grade has been subtracted. Likewise the slope of the material velocity line must be determined after the percentage of the grade has been added to the coefficient of friction.

This development demonstrates the versatility of fundamental mechanisms in fulfilling today's design requirements. Two concepts applied in the shaker drive may be generalized and perhaps adopted in many design situations: (1) Aim for desired or preselected kinematic properties in high-speed or high-load equipment, and (2) Combine mechanism forms to obtain more prolific results than possible with any one of them alone.

Fig. 9—Comparison of performances obtained with the original linkage system and with the combined simpler linkage and irregular gears. Based on 77 9-inch strokes per min and 0.4 coefficient of material friction



They Say . . .

"Industrial research is literally a problem child—it is born of problems. No problems, no research! You begin with a problem—you are searching for facts, for new knowledge in a particular field, an improvement in a product or process, or a product with new usefulness. You think about the problem, you try something and you observe (or measure) what happens. Then you think about what you have observed and try again, and again and again. The research method is simply following the trail of stepping stones of Think and Try and See that leads from problem to answer. You must think through your problem, you must try something, make some experiment, try some new approach; you must see what happens and then think again, on and on and on—and so new miracles are born."—HANS ERNST, *director of research, Cincinnati Milling Machine Co.*

"The faculty for reducing apparently complicated situations to their basic, essential elements is a form of wisdom that must usually be derived from experience . . . It should be our practice to integrate, condense, summarize and simplify facts rather than to expand, ramify, complicate and disintegrate them."—LEWIS A. VINCENT, *general manager, National Board of Fire Underwriters.*

"On the nontechnical front, the mechanical engineer of today must work ever more closely with the social scientists. This is particularly true in the fields of economics, psychology, and employee and public relations, because he frequently finds himself moving into the responsibilities of the management of industrial enterprises."—EVERETT S. LEE, *editor, General Electric Review.*

Hydraulic Control Systems

Basic design of machine hydraulic systems is put on a methodical and consistent foundation in this series of articles to aid development of circuits for maximum simplicity and effectiveness

Part 4—Automatic-Sequence Systems

By R. Hadekel
Consulting Engineer
London, England

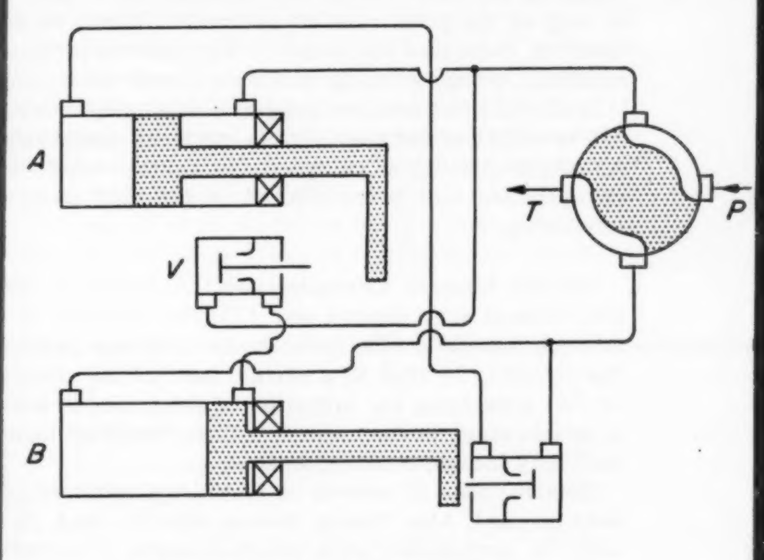


Fig. 104 — Aircraft undercarriage hydraulic system exemplifying a simple half-cycle circuit

MANY hydraulically operated machines are required to perform a number of operations automatically in a predetermined sequence. Generally, there are four main types of sequence operations: half-cycle, single-cycle, multiple-cycle, and continuous operation.

In *half-cycle* operation a single action on the part of the operator initiates operations *A, B, C . . . N* in that order, after which the machine stops. After the operation *N* the system is *not* in its starting position. Another action on the part of the operator initiates the converse sequence. An example is operation of aircraft undercarriage and undercarriage doors. Unloading may be required at the end of the half cycle.

With *single-cycle* actuation a single action on the part of the operator initiates operations *A, B, C, . . . N* in that order, the machine being back at its starting position after operation *N*. For example, in an injection molding machine the cycle is: close dies, inject, retract injector, open dies. Unloading may be required at the end of the cycle.

In *multiple-cycle* actuation a single action on the part of the operator initiates operations *A, B, C . . . N* in that order, and then (without further action on the part of the operator) *A, B, C . . . N* again, and so on for *r* cycles, the machine stopping automatically at the end of cycle *r* when unloading may be required. A typical example is a marking machine in which a roller die must be rolled over the work a given number of times. Operation *N* may be an indexing operation as in a machine for wheel riveting where the wheel must be indexed after each to-and-fro motion of the riveting head, the machine stopping automatically when the wheel has been indexed completely around.

With *continuous operation* a single action on the part of the operator initiates operations *A, B, C . . . N* in that order, and then (without further action on the part of the operator) *A, B, C . . . N* again, and so on indefinitely until the machine is stopped by another action of the operator, after which unloading

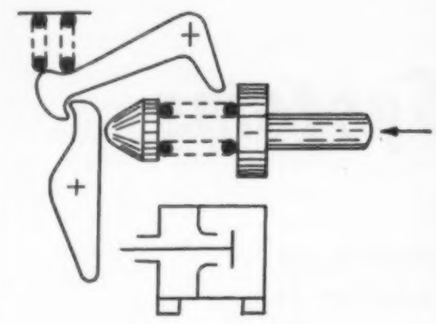


Fig. 105 — Latch - operated trigger type sequence valve with high positional accuracy

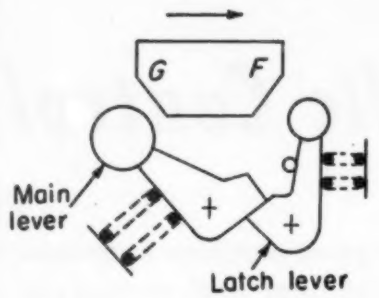


Fig. 106—Latch valve operator employing spring main lever actuation

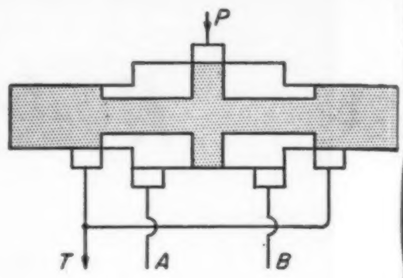


Fig. 107—High-accuracy hydraulic valve with trigger firing action

may be required. Most often, when the operator performs the stopping action the machine is required to complete the cycle before it stops. Starting and stopping actions very often take the form of pressing and releasing a pedal, the machine continuing to operate as long as the pedal remains depressed. *N* may be an indexing operation as found in any general-purpose automatic press, grinding machine, shaper etc.

In any of these systems a definite dwell may or may not be required between certain pairs of operations. Automatic alteration of speed during the course of an operation will be considered in the next part of this series.

Control Element Characteristics: A sequence control element may depend on: (1) The position of a machine member, (2) pressure in a certain part of the circuit, (3) flow in a certain part of the circuit, or (4) time from the initiation of the cycle, or from a certain stage in the cycle. The term "machine member" may include counting devices.

Combinations of several of the foregoing elements may be used. Also, timing devices *must* be used, possibly in conjunction with other elements, if definite periods of dwell are required.

Except in the case of timing devices, there is a general problem in all sequence elements, arising from the following fact: when a sequence element operates, a change is produced in the circuit, and almost invariably the effect of this change is to annul the cause—motion, pressure rise, etc.—which operates the sequence device. Now the latter must usually have a definite motion and usually begins to be operative before this motion is completed. Thus, there is a tendency for the sequence element to be operated only partially, or for an insufficient length of time. In many cases, inertia is sufficient to ensure satisfactory operation or else other methods may be used to introduce a slight delay in the action of the sequence device. In other cases the sequence element, or the valve which it controls, may be given a snap or trigger action.

The problem is usually easier if the sequence elements are of the pilot valve type, since the main valves which these control will usually operate fully even if the pilot elements are only partially operated. Besides, the use of pilot elements automatically introduces a slight delay. With electric elements the problem is easier still, as the travel required is often minute while snap action is often inherent in the device.

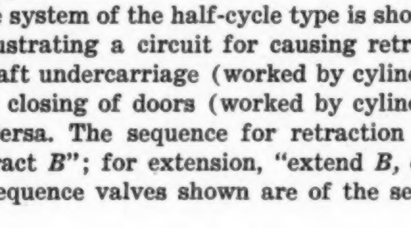
This problem will be dealt with in more detail, but it is mentioned at this stage because of its generality.

Positional Devices: Any nonautomatic valve may be mechanically operated to give positional sequence control. It is also possible to use electric position switches or limit switches which, in turn, control electrically operated valves.

Positional devices are not generally suitable, unless time delay is introduced, where a cylinder must exert full load against internal or external stops (or equivalent) before the next operation is initiated. Thus devices of this type are hardly suitable for initiating automatic return in many kinds of presses, although quite suitable in most cases for initiating a working stroke after the press has returned.

A very simple system of the half-cycle type is shown in *Fig.* 104, illustrating a circuit for causing retraction of an aircraft undercarriage (worked by cylinder *A*) followed by closing of doors (worked by cylinder *B*), and vice versa. The sequence for retraction is, "retract *A*, retract *B*"; for extension, "extend *B*, extend *A*". The sequence valves shown are of the seat-

Fig. 108 — Toggle lever arrangement useful for snap-action valve actuation



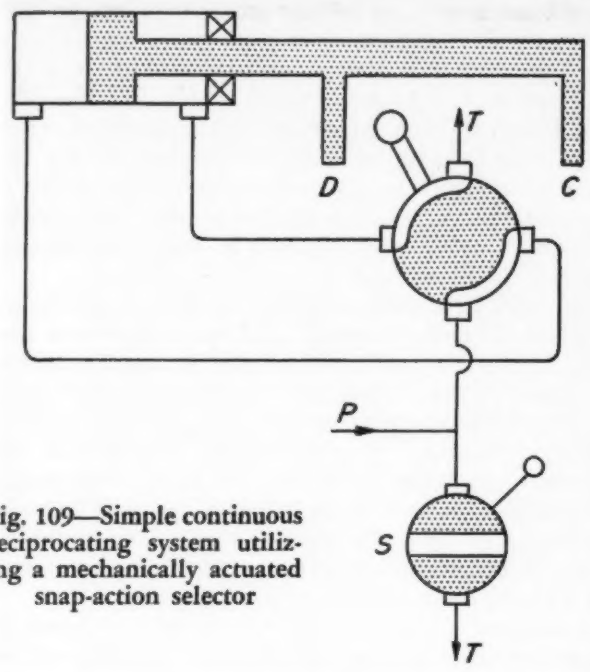


Fig. 109—Simple continuous reciprocating system utilizing a mechanically actuated snap-action selector

ing type, allowing free flow in one direction, which may be helpful in certain cases and at least can do no harm in the particular circuit shown.

This circuit illustrates the type of problem referred to previously. If the pressure required to begin the retraction of *B* is less than that required to complete the retraction of *A*, the latter will cease to move (unless carried on by inertia) when valve *V* is only partially open, causing throttling of flow. Furthermore, the position at which cylinder *A* stops is more or less indeterminate within the limits of travel of the valve. These two effects may of course be tolerated in many cases since, in any event, at the end of the half-cycle both cylinders and the valve will complete their respective travels.

In *Fig.* 105 is shown a trigger type sequence valve in which the initial actuation compresses a valve stem spring through a lever, but operation of the valve is prevented by a latch until a position is reached when the latter is released, the spring then opening the valve to the full extent. Valves of this type may be made to operate with a very high degree of positional accuracy.

A different type is shown in *Fig.* 106. Here the face *F* of the cam operates the latch, allowing the main lever to move under the action of a spring. On the return motion face *G* of the cam resets the valve. To give satisfactory operation, the main lever must be made to travel slightly beyond the latched position during resetting.

Illustrated in *Fig.* 107 is a valve with hydraulic trigger action, for which a "firing point" accuracy of the order of 0.0001-inch is claimed (Ref. 8, Part 1). In theory this valve has a point of equilibrium in the center, but in view of its very high sensitivity this can hardly matter in practice.

A type of snap action useful for many purposes, although not as accurate as trigger action, is shown

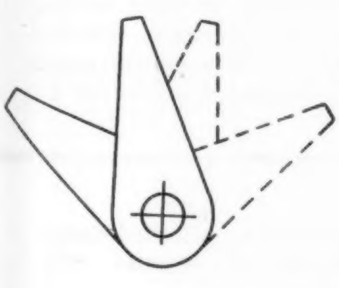
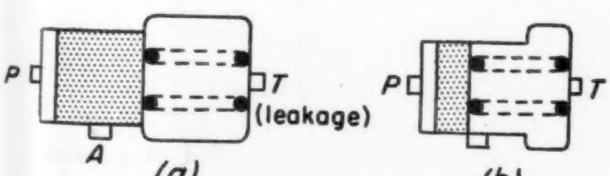


Fig. 110 — Left—Form of pilot valve levers used for reciprocating system control

Fig. 111—Below—Pressure-sequence valves of two-way, *a*, and three-way, *b*, types. Valves remain closed until pressure at the inlet equals the spring load. Design at *b* prevents pressure rise at the outlet before opening



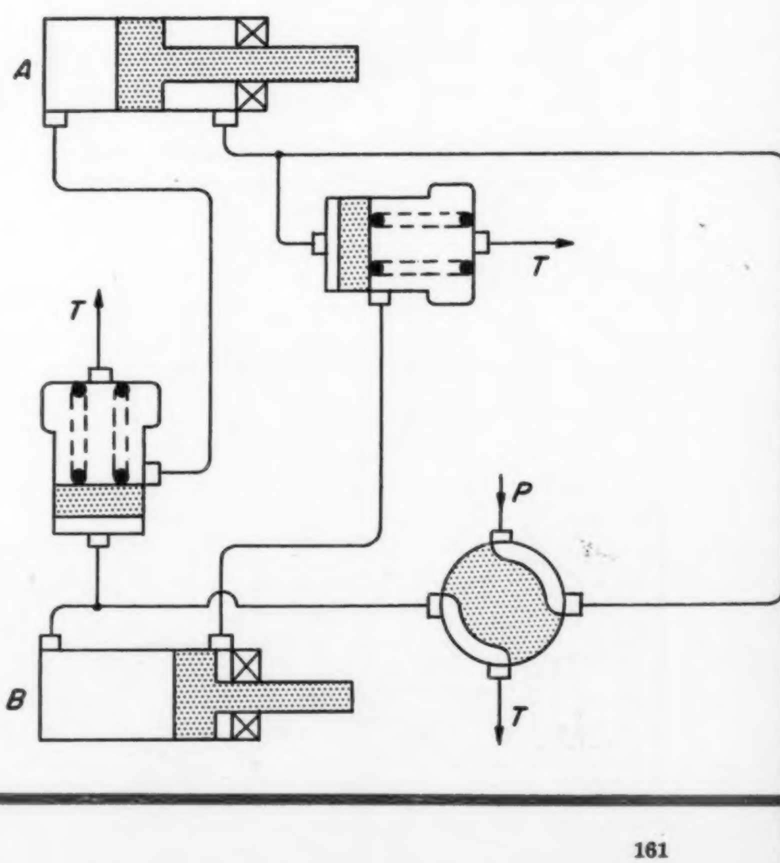


Fig. 112—Right—Equivalent circuit to that of *Fig.* 104 using pressure-sequence valves as shown in *Fig.* 111*b*

in *Fig.* 108. Motion of the machine operates a toggle lever, which is moved further by a cam and spring arrangement. The toggle lever in turn operates the lever actuating the valve, but there is some lost motion between the two, allowing the former to go past center and come under the action of the cam, before the latter reaches its central position (or even before it begins to move). This mechanism may be useful for sequence-operated selectors, worked by direct mechanical action from the machine.

Positional sequence elements are very useful in applications, such as machine tools, where the stroke of a cylinder may have to be varied. This can be done easily by moving the stop or cam operating the valve. *Fig.* 109 illustrates a simple continuous system of this type where reciprocating motion of a machine slide is obtained by a mechanically operated selector which could have the type of snap action shown in *Fig.* 108. Stop *C* reverses the selector at the left-hand end of the stroke, stop *D* at the righ-hand end. The machine is started and stopped by cock *S*, which unloads the pump.

Perhaps more frequently in this type of system the selector is worked from a pilot valve, in which case snap action on the latter is not usually considered necessary. Pilot valves for this type of application usually have stay-put action and are most conveniently made with a lever in the form of a vee with offset arms, as shown in *Fig.* 110, one arm corresponding to each stop, the two stops being of course in different planes. This prevents fouling, and allows unlimited position adjustment. Systems of this type are common on machine tools, particularly grinders.

Positional sequence devices operated by adjustable cams are also useful in determining stopping positions for machine slides, whether or not these have automatic sequence in the usual sense. For instance in rapidly moving machines stopping by hydraulic means may be preferable to direct mechanical action as it reduces impact loads. Gradual stopping can be achieved by causing the machine to actuate a valve giving gradual throttling. If it is preferable to use a pilot valve, the same effect can be obtained by the potentiometer system of *Fig.* 56*b*, Part 1.

Pressure and Flow Sequence Devices: Strictly speaking, flow cannot operate anything and the general principle of so-called flow-sensitive devices is the utilization of the pressure difference created by flow through a resistance. Thus pressure-operated devices are those which utilize changes of pressure, directly available from the operation of the system, while flow-operated devices utilize pressures generated by

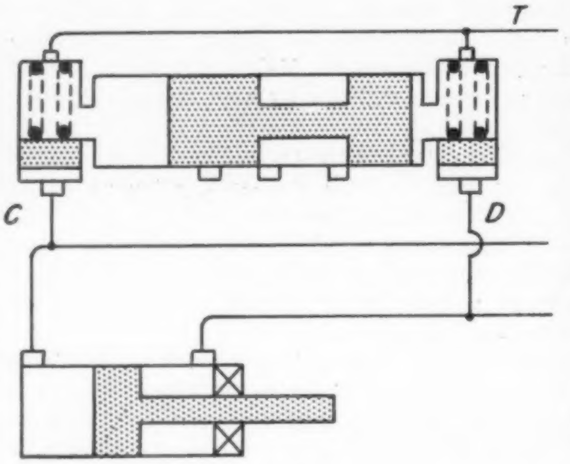


Fig. 113 — Above — Pressure-sequence valve which reverses at completion of forward and reverse movement of a cylinder

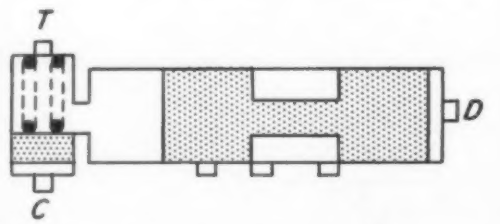


Fig. 114 — Above — Pressure-sequence valve which can be made to reset at any point in the sequence

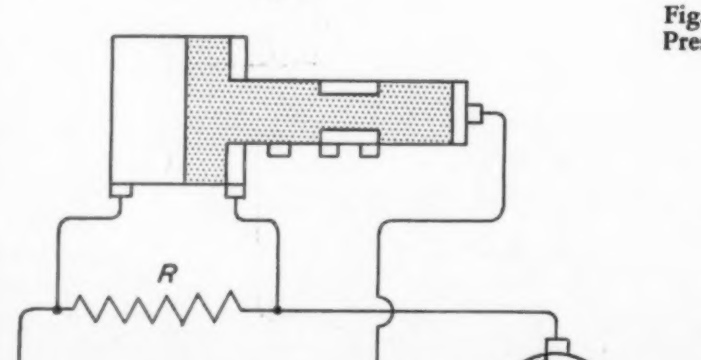


Fig. 115—Left—Flow-sensitive valve controlled circuit. Pressure drop across resistance *R* actuates valve

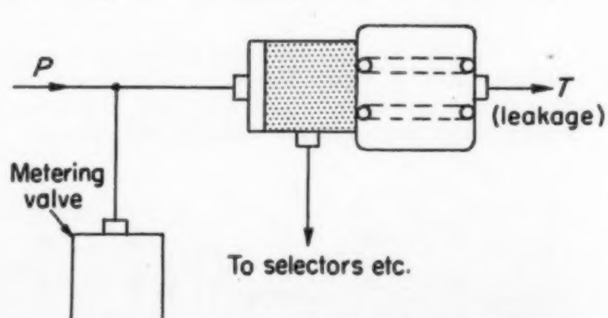


Fig. 116—Use of a metering valve permits the hydraulic control of timing in certain portions of a circuit

resistances deliberately introduced for that purpose.

Unless a cylinder is automatically returned by a positional device, the motion eventually stops due to the cylinder reaching its own limit of travel or sooner if resistance to motion rises sufficiently, and the relief valve or other pressure-limiting device operates. Thus, simultaneously, a rise in pressure up to the relief valve setting and a cessation of flow anywhere downstream of the relief valve branch are obtained. This rise in pressure or cessation of flow, or the two together, may be utilized to initiate the next operation in the sequence.

The usual control element in pressure-operated (not flow) automatic-sequence systems is the pressure-sequence valve, already encountered in earlier articles in connection with pressure maintenance. Both the two-way type of *Fig.* 111*a* and the three-way type of *Fig.* 111*b* or their seating valve equivalents, or indeed any other form with the same characteristics, are useful. Obviously the operating pressure must be higher than the maximum value required to overcome external loads during cylinder motion, but it must be lower than the setting of the relief valve. Hence, there must be a fair margin between the maximum pressure required to move the cylinder and the relief valve setting. The latter, and the pump rating, must be higher than they would with other methods of control. This is one disadvantage of sequence control by pressure.

The two types of valve referred to are obviously not the only ones possible on the same principle and any combination of connections may be established. *Fig.* 112 shows a circuit equivalent to that of *Fig.* 104, but using pressure instead of positional-sequence valves, of the type of *Fig.* 111*b*. This type of system is likely to involve severe power dissipation through the sequence valves.

A characteristic of the pressure-sequence valves so far described is that they reset themselves as soon as pressure drops, i.e., as soon as an alternative path other than through the valve is open to the pump output, provided of course the resistance of that path is not excessive. A different type of pressure-sequence device is shown in *Fig.* 113. Here two pilot pressure-sequence valves are used to operate a third valve of

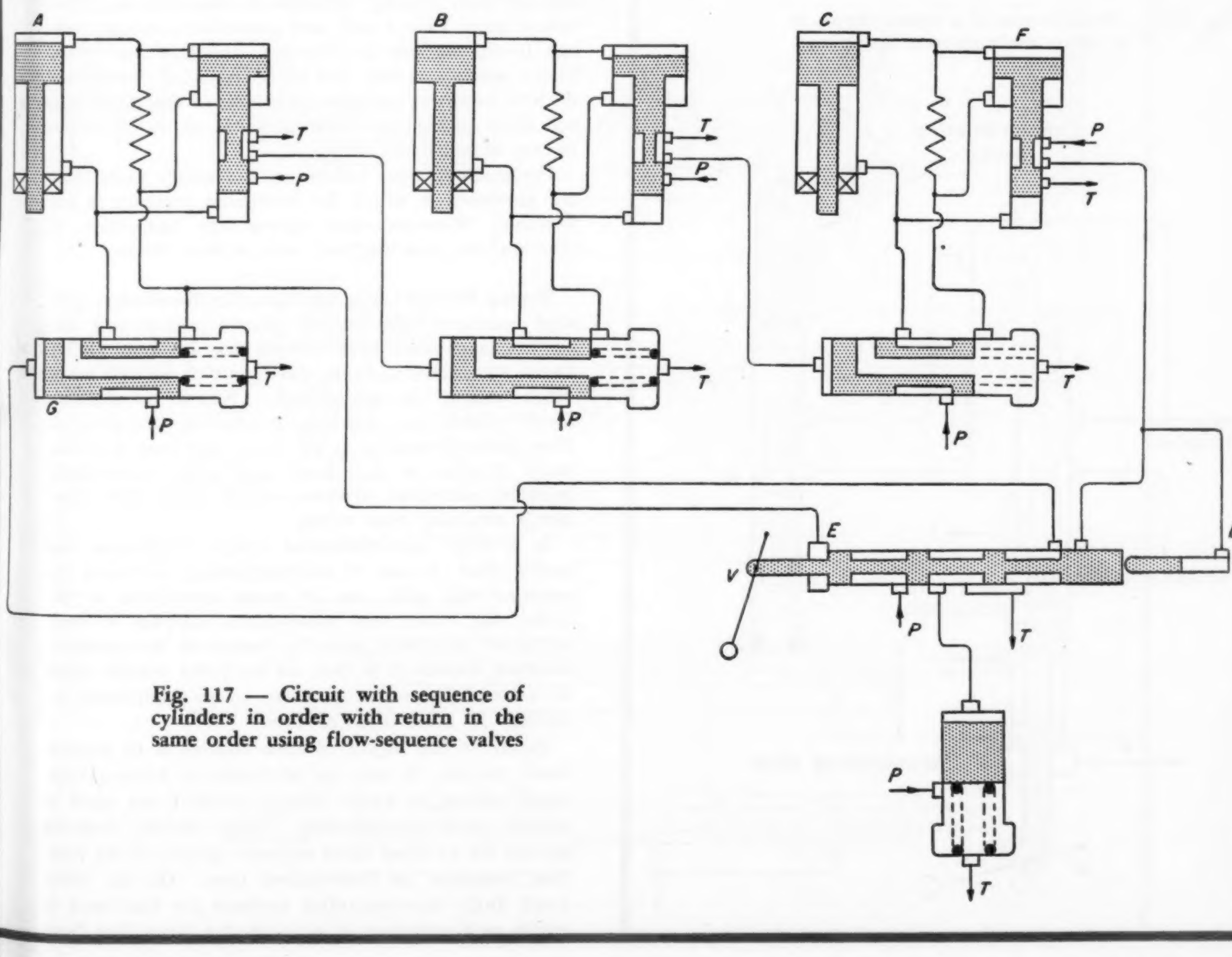
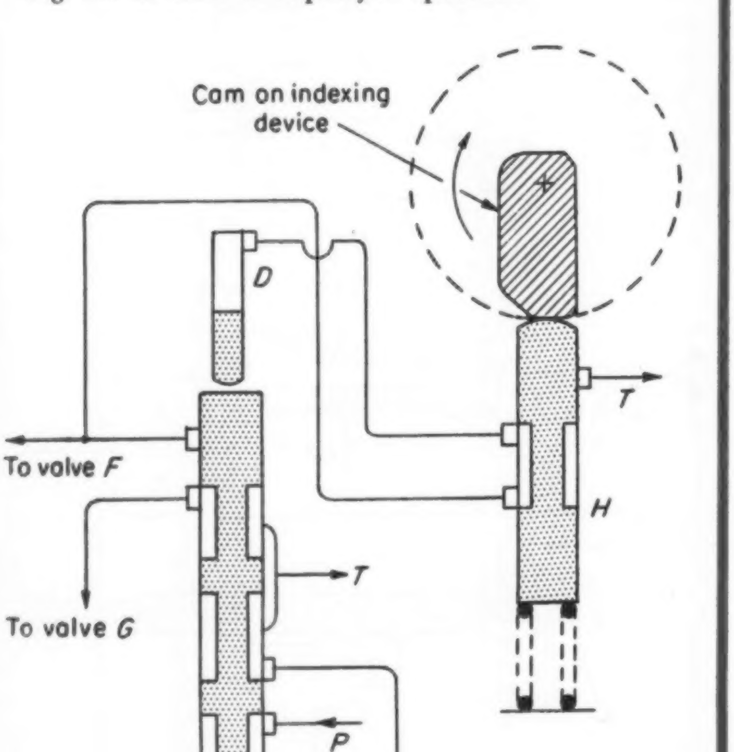


Fig. 117 — Circuit with sequence of cylinders in order with return in the same order using flow-sequence valves

the stay-put type, which may or may not itself be a pilot valve with any combination of connections. When connected as shown, the valve will reverse at the completion of forward and reverse movement of the cylinder. Alternatively, connection *D* may be connected to some other part of the circuit, and the valve may then be made to reset (i.e. move to the left) at any point in the sequence, provided *C* is no longer connected to pressure. Again, in such a case it is not always necessary to have two pilot-sequence valves, and the valve may be as in *Fig.* 114, *D* being connected to some appropriate part of the circuit. The use of valves of these types will be described in subsequent sections.

An example of a flow-sensitive device is shown in *Fig.* 115. The valve is controlled by a relatively large piston acted upon by the pressure drop across the resistance *R* which may be formed by a hole in the piston, or may be external, while the absolute pressure from the opposite line acts on the relatively small area of the valve spool. Proportions are such that the load due to pressure drop in the resistance always predominates over the load due to absolute pressure, even when the latter is at its maximum value. Hence, when

Fig. 118 — Modification of a circuit shown in *Fig.* 117 to obtain multiple-cycle operation



the cylinder is extending the valve will be kept to the left until flow stops, at which point pressure acting effectively on the spool area will push it to the right. When the cylinder is retracting the valve will be kept to the right until flow stops, when pressure will push the valve to the left.

Obviously, any combination of connections can be established by the valve although it is shown as having three ports of unspecified polarities. This type of element has the useful property of staying put until reversed. A further useful property is that the device is not "marginal" in the same sense as the pressure-sequence valve. It can be so proportioned as to operate over quite an appreciable range of flows and pressures. The resistance is set so as not to give too high a pressure loss under the maximum flow to be allowed for and the proportions are then determined by the ratio of minimum flow to maximum pressure.

Within their proper spheres of application, pressure and flow-operated devices may often be preferred to positional devices for the following reasons: First, they usually involve loads strictly coaxial with the valve motion, avoiding the parasitic reaction components which are bound to occur with mechanically operated valves if the latter are actuated by levers or cams and are thus practically free from wear; second, they can usually be made to rely only on close fits for their sealing, whereas mechanically operated valves must have a soft seal somewhere unless leakage to the outside is tolerated, and this seal eventually wears out and has to be replaced. Positional devices, however, are safer in the sense that they cannot allow premature commencement of an operation in case of malfunctioning.

Pressure-sequence valves are obviously unsuitable for machines in which the maximum pressure is adjustable. Flow-sequence valves are unsuitable if flow and/or pressure vary over a wide range.

Timing Devices: In a number of hydraulically operated machines—particularly plastic presses and die-casting machines—dwell periods may be required between successive motions, during which periods loads must usually be maintained. Where no definite dwell periods are required, a position, pressure, or flow-operated system is no doubt the best solution, since systems of that type—and more particularly position controlled systems—entail very little danger if anything goes wrong.

In a fully time-controlled system conditions can easily arise (in case of malfunctioning) in which the machine will omit one or more operations in the cycle, with consequent possibility of damage to itself or to the work and, possibly, danger to the operator. Another drawback is that an adequate margin must be allowed for each operation to be completed, resulting in some loss of time.

Hence, if the object of time control is to provide dwell periods, it may be advisable to have a combined system in which timing elements are used to control dwell periods only. These timing elements are set off by some other sequence device, of the position, pressure, or flow-control type. On the other hand, fully time-controlled systems are also used in which each operation is initiated at a given time from

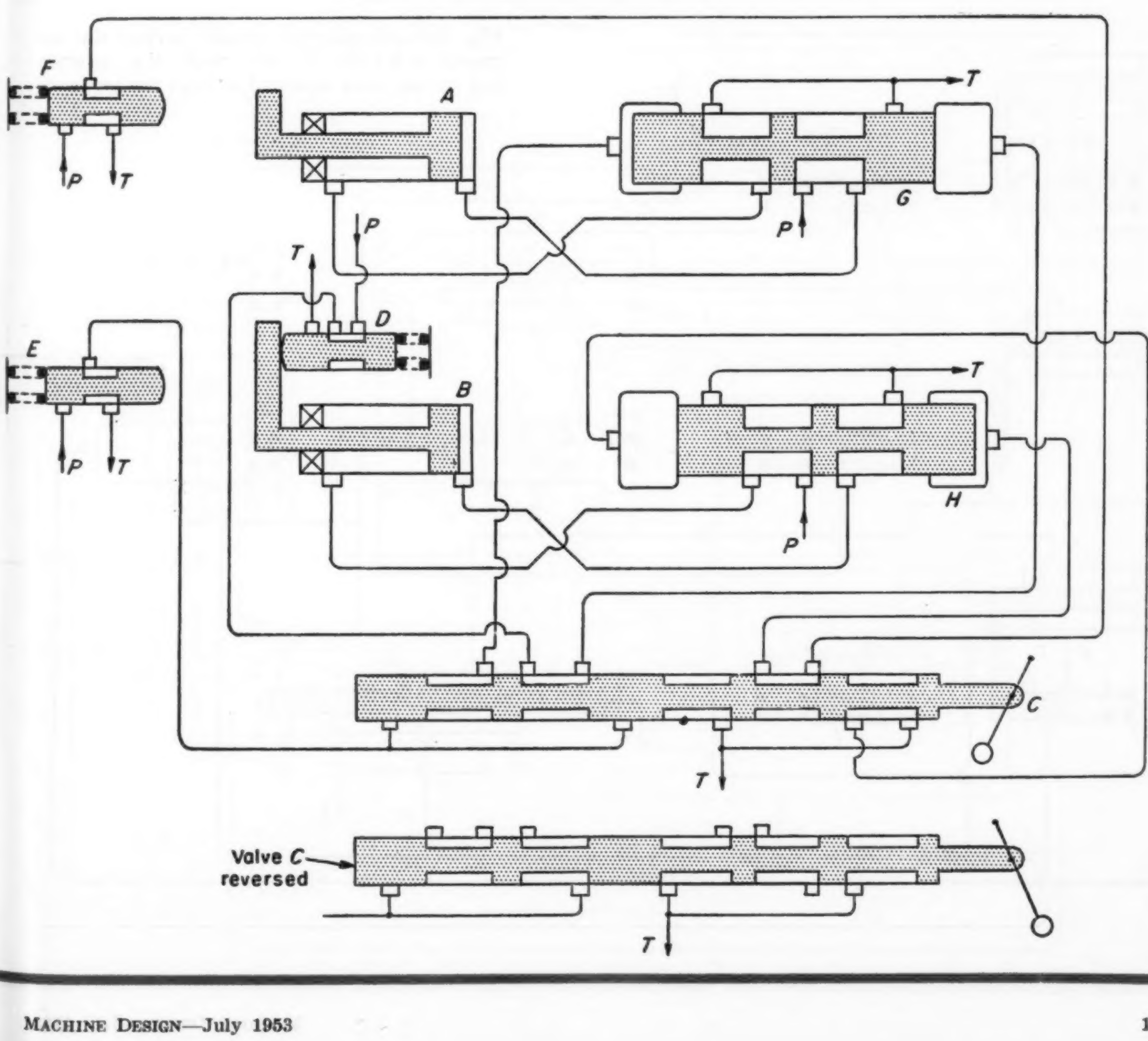
the commencement of the cycle, as determined by some independent timing device and not by the speed of operation of the cylinders. Timing devices, however, are a convenient solution for certain difficult sequences as will be discussed later.

The most usual and no doubt the easiest method of time control is by electric time switches controlling electrically operated valves. If sequence elements proper are to be used as well, they could take the form of pressure switches, which would be equivalent to pressure sequence valves. Like the latter, pressure switches would in most cases reset immediately after operating, but this can be counteracted by causing them to control relays with a self-holding line which can be broken by another pressure switch or by a time switch, as desired. It is also possible to visualize a system using mechanically operated valves worked from a camshaft which is driven through reduction gearing by a synchronous electric motor, or even an induction motor, which would be accurate enough for most purposes if running at

light load. Such a system would really be suitable only for fully time-controlled cycles, and not for time control of dwell periods only. Suitable electric circuits are easily devised for all cases.

Hydraulic control of timing is not out of the question. It may be obtained for instance by tapping a proportion of the pump flow for time control, as shown in *Fig.* 116. The pressure-sequence valve ensures a maintained minimum pressure to the timing circuit, the flow in which is accurately kept to a constant value by a metering valve. The timing flow—normally quite small—may be caused to operate small cylinders that take a strictly fixed length of time to complete their travel, at the end of which they operate some form of valve. These cylinders are best used to control dwell periods only and may be started by some form of sequence device, with adequate provision for resetting. The resulting

Fig. 119 — Half-cycle circuit for aircraft landing gear operation shown when extension has just been selected



circuits are of course apt to be rather complicated.

Counting Devices: It is possible to visualize a system in which there is a single pressure-sequence valve in the pump delivery line to operate a small cylinder for indexing a ratchet, which would thus move by one step at the end of each operation. The ratchet could drive a camshaft which would operate the selectors and other main valves, the sequence being determined by the shape of the cams. A system of this type would be simple enough, and easily adaptable to any type of cycle, but would in some measure suffer from the same weakness as fully time controlled systems in that upsets in the sequence could occur if anything goes wrong. Again, like timing devices, counting devices may be useful in certain difficult cases.

Elementary Sequence Systems: Excluding the plain pressure-sequence valve, the following main kinds of sequence valves have been discussed so far:

1. The pressure-sequence valve of *Fig.* 113, the flow-operated valve of *Fig.* 115, and also stay-put mechanically operated valves. These valves have the characteristic of functioning at the end of one operation of a cylinder and resetting (i.e., operating in the reverse sense) at the end of the converse operation (return of the same cylinder).
2. Mechanically operated valves of the spring returned type, including the trigger action valves of *Figs.* 105 and 106. These function at the end of one operation, and reset at the beginning or during the course of the converse operation (it is also possible to obtain the same action with pressure or flow-sequence valves).

If the operation consisting of the movement of one cylinder in one direction is denoted by A, the converse operation (return) will be denoted by A^{-1}. Elementary sequences are those of the type—$A\ B\ C\ \ldots\ N{:}A^{-1}\ B^{-1}\ C^{-1}\ \ldots\ N^{-1}$—where sequences consisting of the movement of all cylinders in the machine in one set of directions in a given order are followed by their return *in the same order*.

This type of sequence can always be obtained in perfectly straightforward manner by having one selector for each cylinder. The following possibilities are open:

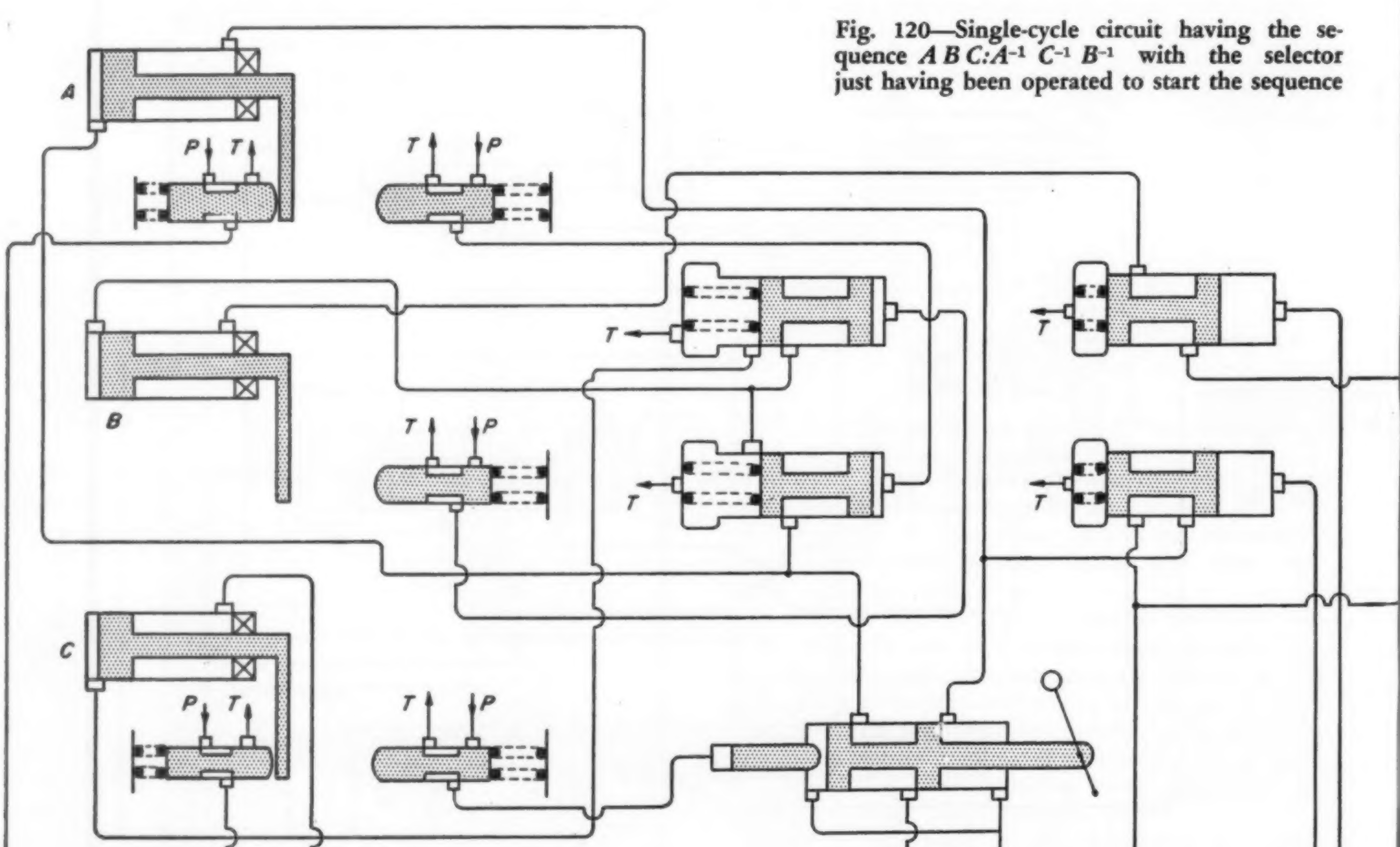


Fig. 120—Single-cycle circuit having the sequence $A\ B\ C{:}A^{-1}\ C^{-1}\ B^{-1}$ with the selector just having been operated to start the sequence

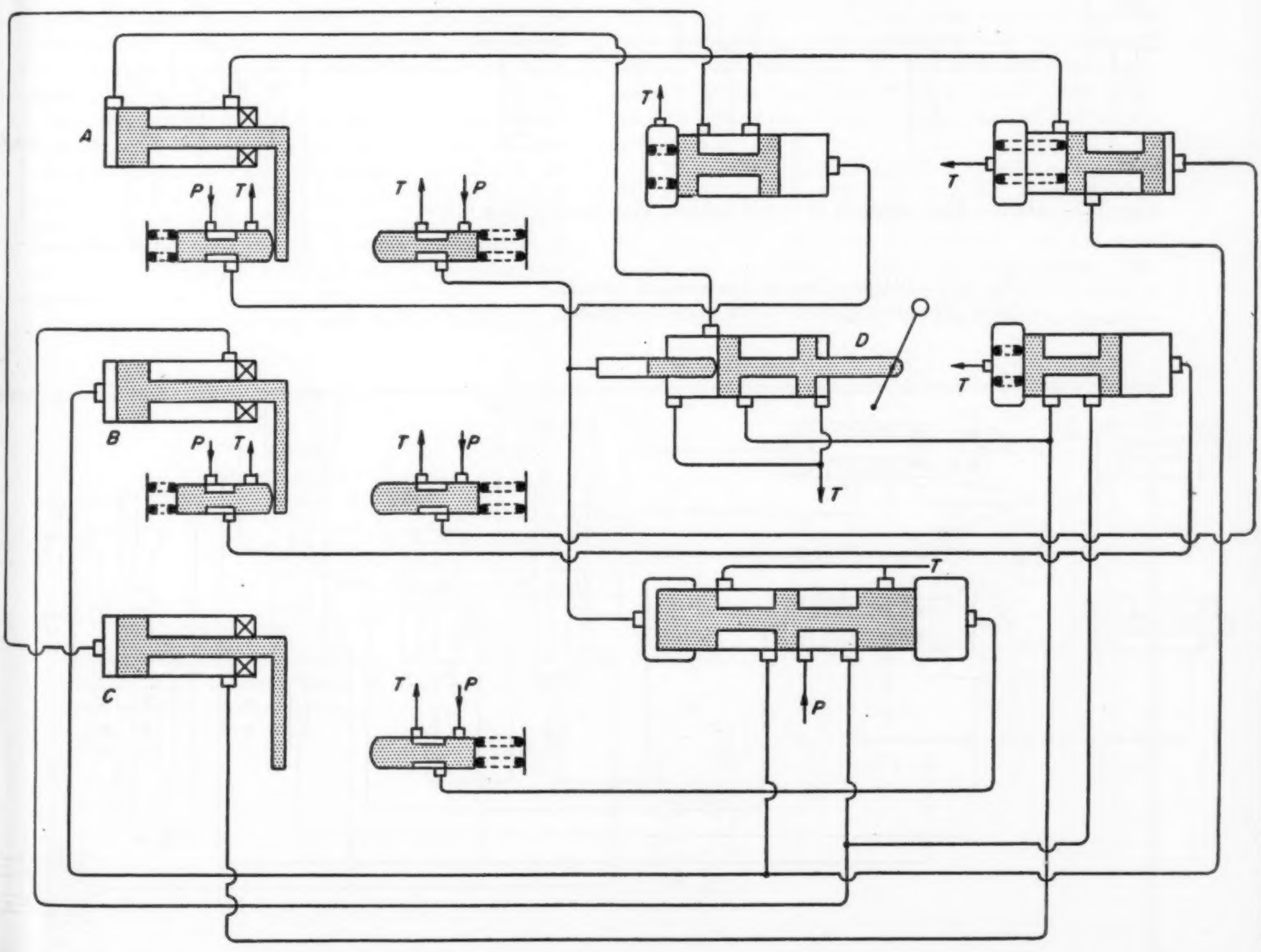
1. Selectors operated mechanically by the cylinders, (cylinder A operates selector b, etc.). Selectors must stay put until reversed when the cylinder reverses and should preferably have toggle action. An elementary case is shown in *Fig.* 109.
2. Selectors operated by pilot valves (of the type indicated in item 2 preceding) at the rate of two pilot valves per cylinder, except for the last one in the cycle which need only have one. Selectors must be of the stay-put type. Alternatively, selectors can be sequence valves of the pressure or flow-operated types of *Figs.* 113 and 115, respectively. In this case some toggle device should preferably be added.
3. Selectors operated by pilot valves of the type indicated in item 1 preceding, one only being required for each cylinder. Selectors may be of stay-put or spring-biased type, the latter requiring fewer connections on the pilot valves.

An example of this kind (sequence $A\ B\ C{:}A^{-1}\ B^{-1}\ C^{-1}$), using flow-sequence valves, is shown in *Fig.*

HYDRAULIC CONTROL SYSTEMS

117. This system is suitable both for single-cycle and for continuous operation. The sequence is initiated by moving valve V right, closing the unloading valve; pressure then comes on connection E through selector G and keeps valve V in right position, even though for the first half of the cycle pressure is also applied at D, tending to reset the valve. After operation C, pressure is removed from both D and E, and valve V still stays put until reset at the end of the cycle when pressure is again applied at D (the pressure required to reset valve V must be less than that required to operate the selector G, otherwise resetting could not be obtained). When the valve resets, the sequence connection from valve F to selector G is broken, and the pump is unloaded. Continuous operation is obtained by keeping valve V operated (say by maintaining pressure on a pedal) and when V

Fig. 121—Automatic circuit for the sequence $A/B\ A^{-1}\ C/B^{-1}\ C^{-1}$ shown in the state reached at the termination of the cycle



is released, the machine continues to operate until the end of the cycle is reached.

A modification to the above circuit, ***Fig.* 118**, obtains multiple cycle operation. Valve *V* is the same as before, but the resetting connection to *D* is interrupted by valve *H* until the end of the series of cycles is reached. Valve *V* must be held until the first indexing operation.

Exactly the same circuit can be used with pressure-sequence valves of the types of ***Fig.* 113**, or with mechanically operated pilot-sequence valves having stay-put action reversed at each end of the cylinder travel.

For multiple-cycle operation some form of indexing device must be present even if indexing is not required by the machine itself. In this case, indexing can be effected by a ratchet mechanism directly coupled to one of the selectors. Starting and resetting arrangements may be somewhat simpler if selectors have stay-put instead of spring-biased action.

Similar principles may be applied sometimes in more difficult cases if operation is on a half-cycle basis, provided each half-cycle is relatively simple. An example is the sequence $A\,B\,A^{-1}{:}A\,B^{-1}A^{-1}$ which occurs for instance in certain aircraft when undercarriage doors must be closed both when the under-

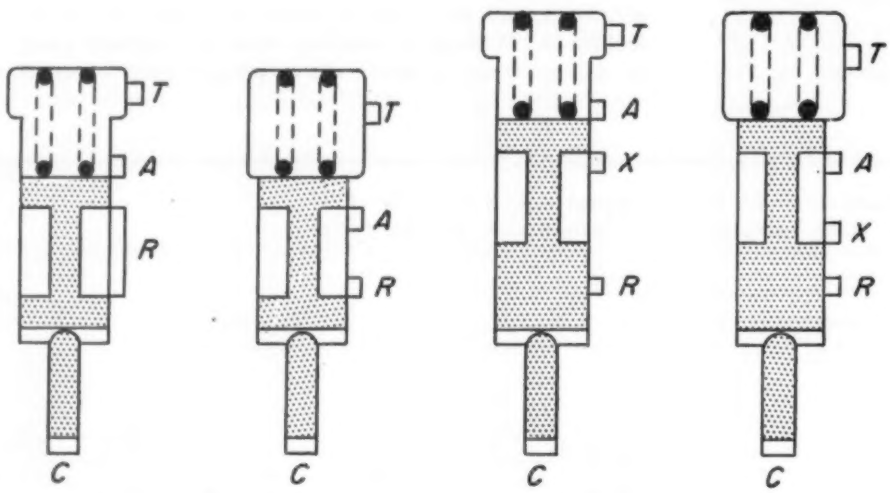


Fig. 122—Above—Four versions of a self-holding pressure-sequence valve

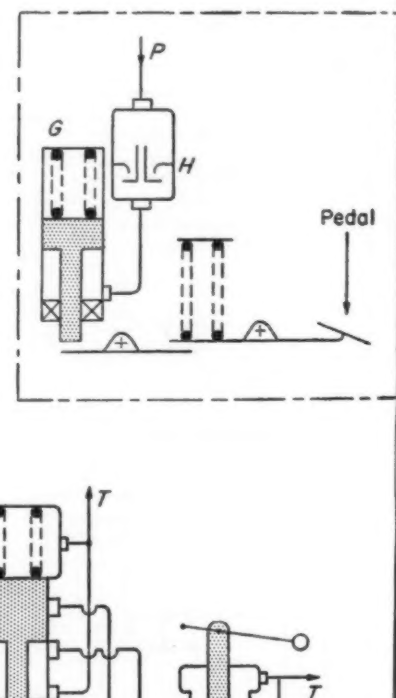
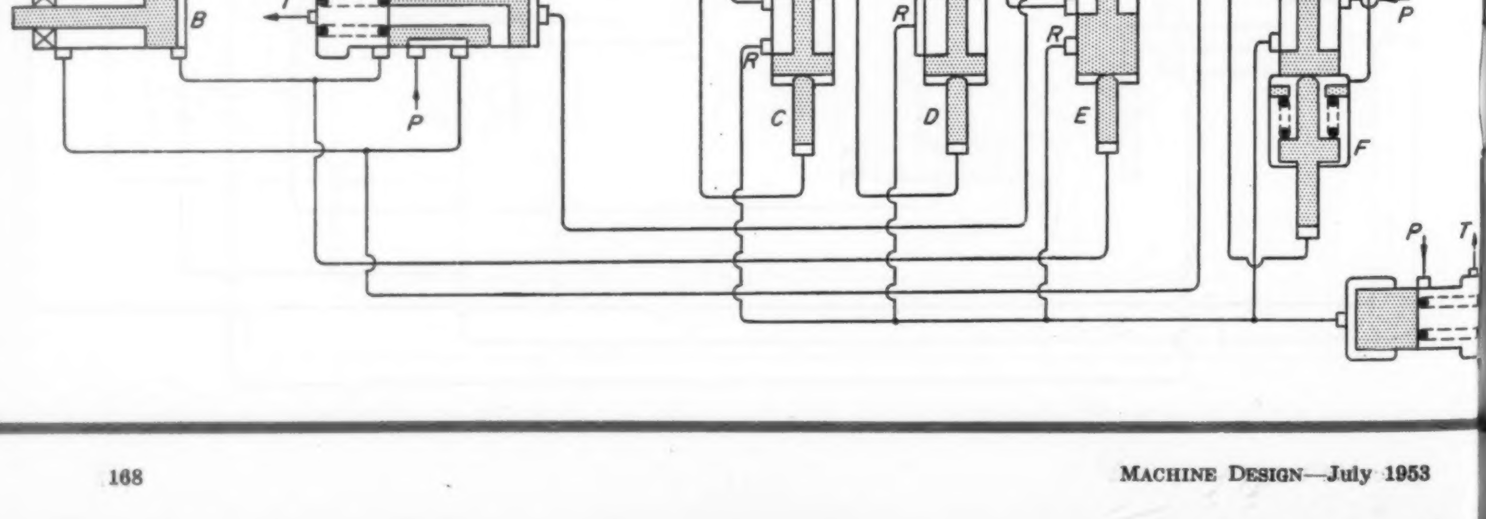


Fig. 123—Below—Circuit composed of several pilot pressure-sequence valves ganged together



carriage is extended and when it is retracted (A is open doors, B is extend undercarriage). A circuit for this is shown in *Fig.* 119 the system being shown when extension has just been selected. Control is effected by the manually operated pilot valve C, which has the effect of switching over the interconnections between the mechanically operated pilot valves, D, E, F, and the selectors, giving different sequences in the two senses.

Series Sequence Systems: Consider a sequence in which all cylinders move forward in a given order, but reverse in a different order, e.g. $A\,B\,C{:}A^{-1}\,C^{-1}\,B^{-1}$. Examples of this kind have been given in *Figs.* 104 and 112 for two cylinders only, and for half-cycle control. The principle of the above circuits can be extended to any number of cylinders and to full cycle systems. Each (except the last to move in each half of the cycle) must operate two full-flow valves that are connected in series with the selector, which must be automatically reversed (excluding the case of half-cycle operation) at the end of the first half of the cycle. As compared with the system of *Fig.* 117 which requires n selectors if there are n cylinders, the present system requires one selector only, but $2n—1$ full size valves altogether.

In the circuit of *Fig.* 104 the sequence valves allow free return, but this is not necessary in that particular case. In the general case, free return is necessary or, alternatively, the sequence valves must be three-way types giving a connection to tank when reset.

Sequence valves may be of any type and, again, pilot-sequence valves may be used, as in the two examples to follow. *Fig.* 120 shows a circuit for the sequence $A\,B\,C{:}A^{-1}\,C^{-1}\,B^{-1}$, the selector having just been operated to start the sequence. Operation is on single-cycle basis, but it is obviously possible to extend the principle to multiple-cycle or continuous operation.

Considering now the sequence $A\,B\,A^{-1}\,C\,B^{-1}\,C^{-1}$ in cyclical order alone (i.e., arranging the operation symbols in a circle), this is really the same as the one just discussed. Writing it $A/B\,A^{-1}\,C/B^{-1}\,C^{-1}$, then the slant line indicates the points at which the selector must be reversed, and both reversals must now be automatic. Starting must be effected by a manually operated valve corresponding to one of the sequence valves in the system of *Fig.* 120. The resulting system is shown in *Fig.* 121, the circuit being in the state reached at the end of the cycle; starting is obtained by moving valve D to the left. This valve can be reset at any convenient point in the cycle, as shown, after operation A^{-1}.

By such methods it is possible to deal with all sequences involving two or three cylinders, provided each does only one to-and-fro motion per cycle, and excluding the cases $A\,A^{-1}\,B\;B^{-1}\,C\,C^{-1}$ and $A\,B\;B^{-1}C\;C^{-1}\,A^{-1}$.

Relay Circuits: As already mentioned, any type of sequence can be obtained by using timing or counting devices. If, however, the circuit is limited to sequence devices proper, which is often convenient and desirable, it seems probable that it is still possible to get any type of sequence by using electric relays with self-holding contacts, or hydraulic valves having similar characteristics, i.e., valves which when operated stay operated until reset at some stage which can be arbitrarily chosen, usually at the end of the sequence.

A manually operated valve of this type has been illustrated in the circuit of *Fig.* 95, (Part 3), the valve being locked in position against a spring (say by a magnet) until reset by pressure. Similar principles are applicable of course to pressure and flow-operated valves, and the action can be incorporated in selectors or in pilot valves; an example of a pressure-

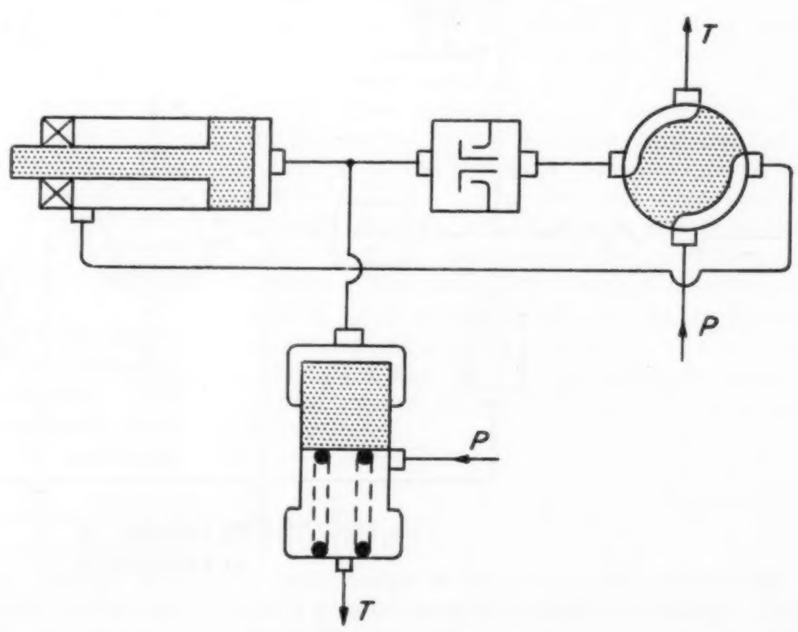


Fig. 124—System for a machine with a single cylinder which effects unloading with the sequence valves

sequence valve of this type has been given in *Fig.* 114.

Fig. 122 illustrates four versions of a type of pressure-sequence valve which holds itself until reset by releasing (not applying) pressure, which is a desirable feature in many cases. Like a self-holding relay, the valve can reset only if the operating connection is no longer to pressure, which can always be arranged to be the case at the required time. When pressure at connection *C* reaches the operating value, the small piston moves the valve against the spring until pressure from connection *R* reaches the bottom of the spool, maintaining the valve operated. The holding pressure need only be comparatively low. After pressure has been removed from connection *C*, the valve will reset when connection *R* is put to tank, or even if *R* is merely blocked off, as in any event a leakage path to tank should be provided (by means of a small hole through the spool) to prevent accumulation of pressure in the reset position.

Several valves of the type of *Fig.* 122 may be ganged together, with a single operating piston as illustrated in the following example. Circuits based on such valves must take into account the fact that the valve will operate not only when the cylinder to which it is coupled completes an operation, but whenever any other cylinder completes an operation, as long as the valve is connected to pressure. The circuit must therefore be arranged to prevent un-

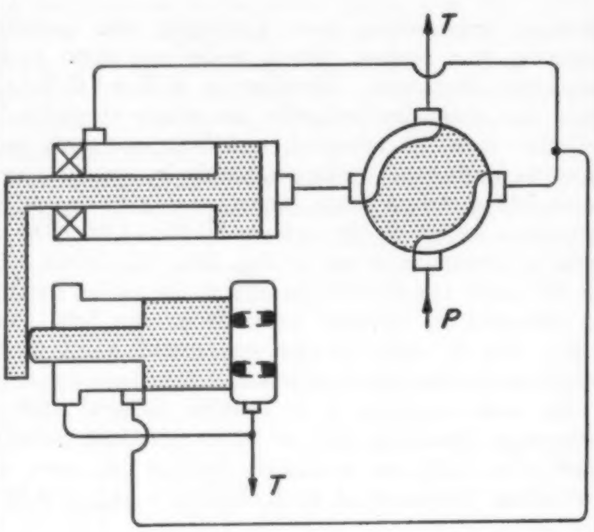
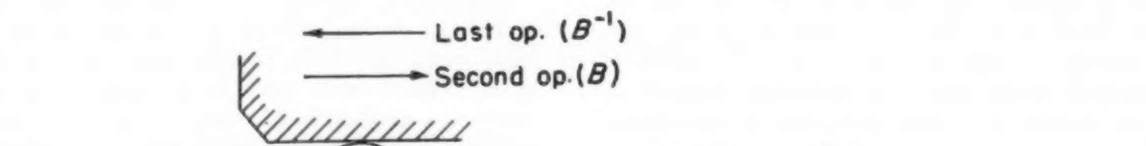


Fig. 125—Simple system, similar to that of *Fig.* 124, which unloads at the end of the cycle

Fig. 126—System of *Fig.* 29, employing pilot-sequence unloading, applied to a more complicated sequence



timely operation or reoperation of valves.

An example of a circuit using pilot valves of this type, in conjunction with spring-biased selectors, is shown in ***Fig.*** **123**, the sequence being $A\,A^{-1}\,B\,B^{-1}$. Valve F is of a different type from the others, being held down (say by a magnet) against a light spring (not shown), and released by pressure-sequence action. The valves C, D and E operate in turn during the sequence and stay operated. At the end of the last operation valve F is reset by pressure and pressure is removed from the R connection on valves C, D and E, which reset. Unloading is obtained at the end of the cycle.

Continuous operation may be obtained for instance by adding the components shown within the chain dotted rectangle in ***Fig.*** **123**. As long as th pedal is depressed, piston G is freed, and during the cycle will be pushed in by pressure. At the end of the cycle the pump is unloaded, pressure drops, and piston G will return, the restriction valve H being arranged to make the return sufficiently slow to allow the valves time to reset. At the end of its return G will operate valve F, and the cycle begins again.

It should be possible by similar methods to obtain even the most complicated sequences, including those involving more than one to-and-fro motion of one or more cylinders, as for instance $A\,B\,A^{-1}\,B^{-1}A\,B\,B^{-1}\,A^{-1}$. In very complicated cases however electric relay control would probably be cheaper.

Sequence Devices for Unloading Control: The methods of unloading described in previous articles are best suited to machines and systems involving a number of independent operations, or at least a number of operations which can be performed in any order. In the present article so far, many examples have been given of unloading in conjunction with automatic-sequence control. Sequence devices are, however, also eminently suitable for effecting unloading in machines which have to perform a definite sequence of operations, even though that sequence may not be achieved by automatic means.

Fig. **124** shows a system applicable to a machine with a single cylinder, say a simple type of riveting press. While the cylinder is extended, or when it is left extended, pressure is available to close the unloading valve and keep it closed. Pressure is still available while the cylinder is being retracted, due to the restriction valve, but when retraction stops the unloading valve control connection is put to tank and the valve opens. At the cost of a few additional complications, the system could be extended to certain more difficult cases.

As already mentioned earlier, automatic open-center systems, and the equivalent pilot and electrical systems of ***Figs.*** **95, 102** and **103**, respectively, lend themselves to sequence control of unloading, by making the latter depend on pressure in one particular line, instead of in the delivery main.

Perhaps the most convenient method in many cases is to use positional devices, which are arranged to unload the machine in its "end of cycle" position, but to be overridden when the first operation of the sequence is selected. The simplest system of this type, equivalent to that of ***Fig.*** **124**, is shown in ***Fig.*** **125**. The unloading valve is mechanically operated by the cylinder at the end of its retraction stroke, but becomes inoperative when the selector is moved to the extend position. Here, again, the problem of partial opening of the valve is met (unless the latter has trigger or snap action), but the valve will at least open to the extent required to reduce pressure to the value just necessary to move the cylinder at the end of the retraction stroke, which may well be low enough.

Fig. **29** (Part 1) shows a solution using a pilot-sequence valve controlling an unloading valve, which avoids the problem of incomplete opening. The rotary valves shown may be main selectors, or may be pilot valves controlling the main selectors; the valves are shown in their initial position. The sequence (not automatic) is $A\,B\,B^{-1}\,A^{-1}$. The circuit also illustrates the application of a safety gate unloading device, operated by pilot valve E, while normal unloading is controlled by pilot valve F. Note that a greater degree of safety would be obtainable by using the safety gate pilot valve in the reverse sense, i.e., allowing it to open when (and only when) the gate is closed. There would then be no danger of the gate becoming ineffective if the valve sticks.

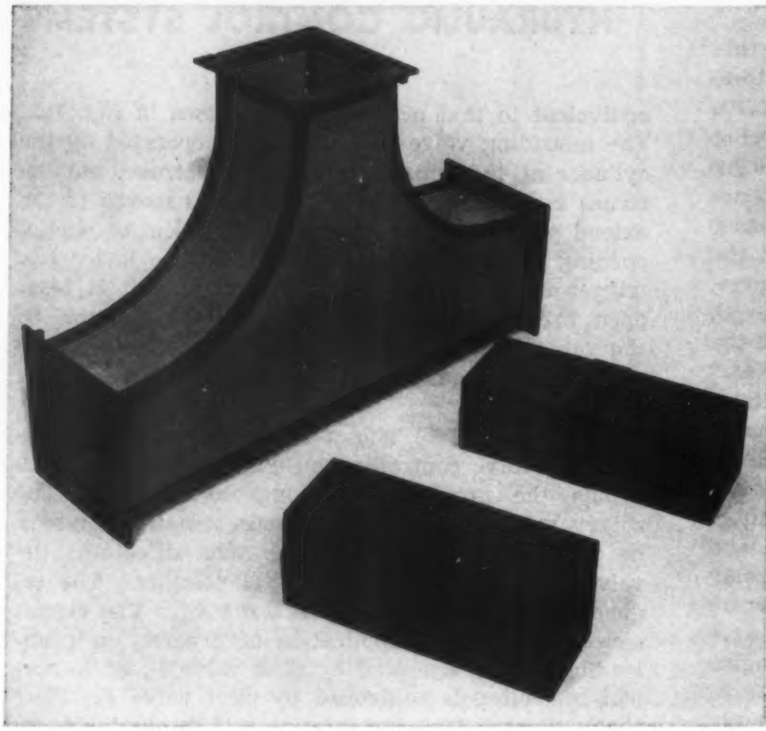
Note also that in this and many other cases, spring-returned positional sequence valves are best operated by cams (as shown) instead of stops, i.e., by members moving at right angles to the direction of motion of the valve, thus ensuring full operation without relying on an exact end position, and preventing damage in the event of overtravel.

The system of ***Fig.*** **125** is applicable to more complex sequences with directly controlled selectors and the system of ***Fig.*** **29** (Part 1), either with direct or pilot-controlled selectors, always provided the last operation is the converse of the first—the sequence is of the type $AB\ .\ .\ .\ A^{-1}$. ***Fig.*** **126** illustrates a slightly more complicated system, as applied to the sequence $AB\,A^{-1}\,B^{-1}$. Here, again, the rotary valves are shown in their initial position, and may be main selectors or pilot valves. The system of ***Fig.*** **126** is applicable to any sequence provided the initial position is not repeated until the end of the cycle; it would not work for a sequence such as $A\,B\,B^{-1}\,A^{-1}\,C\,C^{-1}$. For difficult cases such as the latter, a solution could no doubt still be obtained by using the principles discussed in the section on relay circuits. The unloading systems described in this section could of course also be applied to machines with automatic-sequence control.

Fifth and final part in this series, to appear in the August issue of MACHINE DESIGN, will cover hydraulic servos.

They Say . . .

"Poor design is very much like a rotten egg. Most times it is obvious only after it is broken."—KURT O. TECH, *The Cross Co.*



PLASTIC DUCTING

By H. J. Stark
Bureau of Ships
Department of the Navy
Washington, D. C.

DUCTING fabricated of plastic sheet has several advantages over metal ducting, mainly lightness in weight and resistance to corrosion, particularly important where corrosive fumes or air containing acid vapors is transmitted. Additionally, such plastic ducts can be fabricated by techniques resembling standard sheet-metal manufacturing methods, and thus extensive modifications to adapt present designs are not necessary. For this purpose, requirements are that the plastic be:

1. Bent or fabricated to make circular, flat-oval or rectangular ducts by standard sheet-metal techniques without special equipment
2. Made permanently watertight
3. Fabricated into fittings such as elbows, Y's, bellmouths, and into satisfactory dampers and closures.

Two plastics which have been investigated and seem most satisfactory in these respects are rigid unplasticized polyvinyl chloride sheet (thermoplastic), and fibrous glass-mat filled polyester resin (thermosetting). Melamine and phenolic-resin binders may be used instead of the polyester.

Both can be bent readily, although the polyvinyl chloride takes sharp bends whereas the glass polyester does not. Simple angle shapes, slotted to receive cut sheet, seem to be most satisfactory, although this procedure may not be necessary with polyvinyl chloride.

Permanent water-tightness can be accomplished with both types of plastics by the use of simple adhesives, producing bonds as strong as the material itself. Fittings can be made of the polyvinyl chloride sheet by welding with a hot-air torch, and can be shaped on simple jigs with the application of heat. Bolted flanges and slip sleeves are adaptable with both materials, although adhesives should be used with sleeve joints.

Since the plastics are generally lower in physical strength than aluminum or steel, supports at more frequent intervals will probably be required, especially with ducts for heated air. The thermoplastic polyvinyl chloride is most susceptible to temperature, retaining approximately 50 per cent of its strength at 150 F. Both materials, however, are considered satisfactory for transmission of heated air at 130 to 150 F. Melamine or phenolic-resin binders would probably provide even better heat resistance.

These plastics also have several other advantages: they will not support combustion; they have internal damping characteristics which reduce noise generation from vibration caused by air flow; and they do not have to be insulated to prevent condensation, since plastics are good insulators. Both materials have adequate resistance to shock and impact. A sample study to show relative weight and cost of various ducting materials is shown in TABLE 1.

Table 1—Ducting Material Cost*

Material	Weight (lb)	Cost
Galvanized steel	2.5	$0.175
Aluminum	0.87	0.39
Stainless steel	2.5	1.25
Glass-polyester plastic	0.48	0.29
Polyvinyl chloride plastic	0.48	0.40

*Based on 12 by 12 by 1/16-inch sheet.

Design of
Thin Cantilever Beams

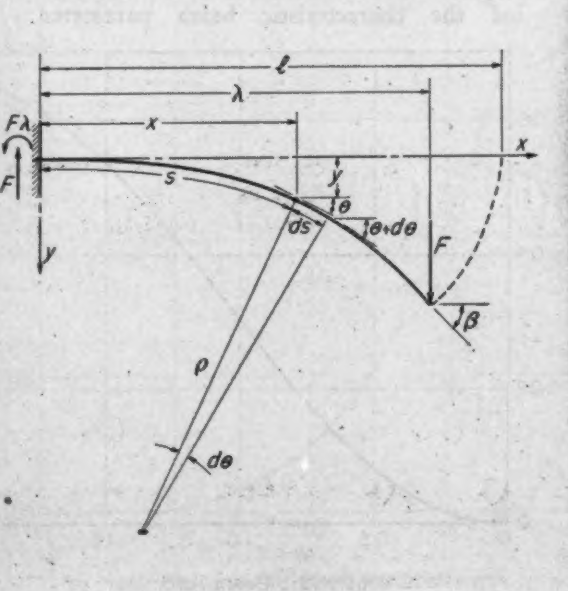
By J. H. Baltrukonis
Technical Staff
Hughes Research & Development Lab.
Culver City, Calif.

DESIGNING a cantilever leaf spring may appear to be a clear-cut application of elementary beam theory. However, extreme caution should be employed in applying this theory if the spring is to be loaded to the extent that large deflections result. Predictions by this simplified theory may become quite inaccurate and in extreme cases will indicate deflections greater than the length of the beam.

Explanation of these erroneous predictions is found in the derivation of the theory. The square of the first derivative in the curvature formula is neglected, and no correction is made for the shortening of the moment arm as the loaded end of the beam deflects. If deflections are small, these simplifications introduce negligible error; however, in the event of large deflections these omissions make the theory very inaccurate.

Therefore, there should be a criterion for indicating whether the elementary beam theory will yield accurate results in a specific case. If this theory is proved inadequate, then an alternate theory—the large deflection theory—should be available. This article presents techniques to supply both of these needs.

Fig. 1—Co-ordinate system and nomenclature for a thin cantilever beam



Large Deflection Theory: The problem of large deflections and their attendant stresses in thin beams has been studied recently by several investigators. Gross and Lehr[1] presented an approximate solution: Barten,[2] and Bisshopp and Drucker[3] obtained exact solutions by similar methods; and Seth[4] used a different approach to arrive at essentially the same result. The solution used in this paper is substantially that due to Barten. The basic premise for the derivation of this theory is the Bernoulli-Euler theorem. It was pointed out by Timoshenko[5] that for relatively thin, wide beams, this theorem must be written

$$M = \frac{D}{\rho} \qquad (1)$$

where M is the bending moment at any point x on the neutral axis of the beam, ρ is the radius of curvature at this point, and D is the flexural rigidity, given by

$$D = \frac{EI}{1-\nu^2} \qquad (2)$$

where E is the modulus of elasticity of the material, I is the area moment of inertia of the cross section, and ν is Poisson's ratio.

Noting *Fig.* 1, Equation 1 can be written

$$\frac{d\theta}{ds} = q^2\,(\lambda - x) \qquad (3)$$

in which λ is the distance from the clamped end to the end force F and

$$q^2 = \frac{F}{D} \qquad (4)$$

Integration of Equation 3 over the entire length l of the beam gives

$$q\,l = K\,(\alpha) - G\,(\alpha, \phi_1) \qquad (5)$$

The notation is similar to that used by Jahnke and

[1] References are tabulated at end of article.

Emde[6] for the complete $K(\alpha)$ and the incomplete $G(\alpha, \phi_1)$, elliptical integrals of the first kind, where

$$\alpha = \sin^{-1} \cos \frac{1}{2}\left(\frac{\pi}{2} - \beta\right) \quad \text{(6)}$$

and

$$\phi_1 = \sin^{-1} \frac{\cos \frac{\pi}{4}}{\cos \frac{1}{2}\left(\frac{\pi}{2} - \beta\right)} \quad \text{(7)}$$

It develops that ql is a characteristic parameter in terms of which all the later results can be represented. Since q can be readily calculated from Equation 4, Equation 5 merely expresses a relationship between the characteristic beam parameter ql and the angle β, which is the angle at the end of the beam between the neutral axes of the deflected and the undeflected beams as shown in *Fig.* 1.

Integration of Equation 3 over the beam from the clamped end to any arbitrary point (x, y) on the elastic curve of the deflected beam results in

$$\frac{x}{l} = \frac{\sqrt{2 \sin \beta} - \sqrt{2 (\sin \beta - \sin \theta)}}{K(\alpha) - G(\alpha, \phi_1)} \quad \text{(8)}$$

and

$$\frac{y}{l} = \frac{G(\alpha, \phi) - G(\alpha, \phi_1) - 2 H(\alpha, \phi) + 2 H(\alpha, \phi_1)}{K(\alpha) - G(\alpha, \phi_1)} \quad \text{(9)}$$

where $H(\alpha, \phi)$ and $H(\alpha, \phi_1)$ denote incomplete elliptical integrals of the second kind with α and ϕ_1 given by Equations 6 and 7 and

$$\phi = \sin^{-1} \frac{\cos \frac{1}{2}\left(\frac{\pi}{2} - \theta\right)}{\cos \frac{1}{2}\left(\frac{\pi}{2} - \beta\right)} \quad \text{(10)}$$

Equations 8 and 9 are a pair of parametric relations for the dimensionless co-ordinates of points on the elastic curve of the deflected beam. From these relations and Equation 5 dimensionless shape curves were plotted as a function of the characteristic beam parameter ql as shown in *Fig.* 2.

Although all necessary information concerning deflections is contained in these dimensionless shape curves, sometimes it is useful to know the maximum deflection of the beam parallel to the y axis. This deflection is that of the beam under the load, that is, for $\theta = \beta$. *Fig.* 3 shows a curve of the dimension less maximum deflection y_{max}/l plotted as a function of the characteristic beam parameter ql.

It can be shown that by the large deflection theory the maximum stress in a beam is given by

$$\sigma_{max} = \frac{Flh \sqrt{2 \sin \beta}}{2\, Iql} \quad \text{(11)}$$

where h is the thickness of the beam. This maximum stress occurs at $x = 0$ and is compressive for $y = +h/2$ and tensile for $y = -h/2$.

Comparison of Theories: Elementary beam theory can be applied with sufficient accuracy to the study of end-loaded cantilever beams that are relatively stiff and where the deflections remain rather small. However, if the beam is flexible and the applied loads are such as to produce large deflections, elementary beam theory may be quite inadequate. The large deflection theory, in addition to its utility for purposes of design, can be applied immediately to determine quantitively the limits of validity of the elementary theory. This can best be done by calculating ratios of corresponding values predicted by the two theories. It probably will be sufficient to compare the values predicted for the maximum deflections parallel to the y axis and for the maximum stresses.

For the elementary beam theory it is safe to assume that the deflections of all points of the beam

Fig. 2—Dimensionless shape curves for points on the deflected beam

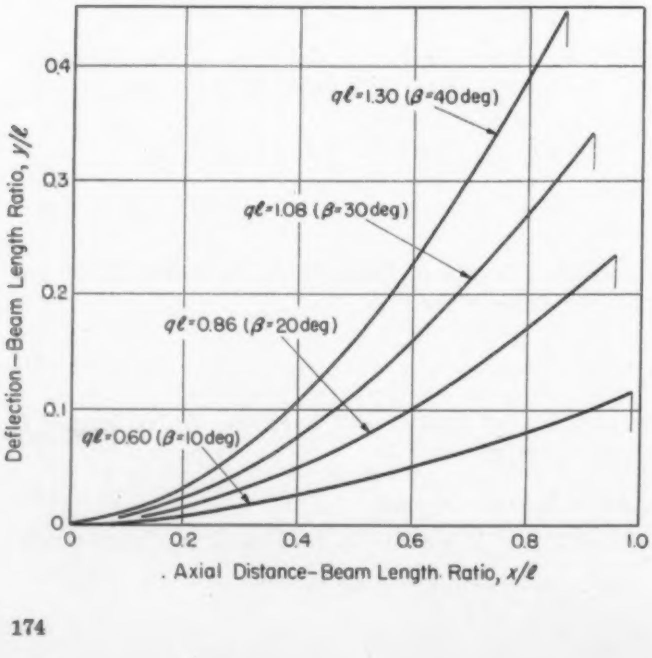
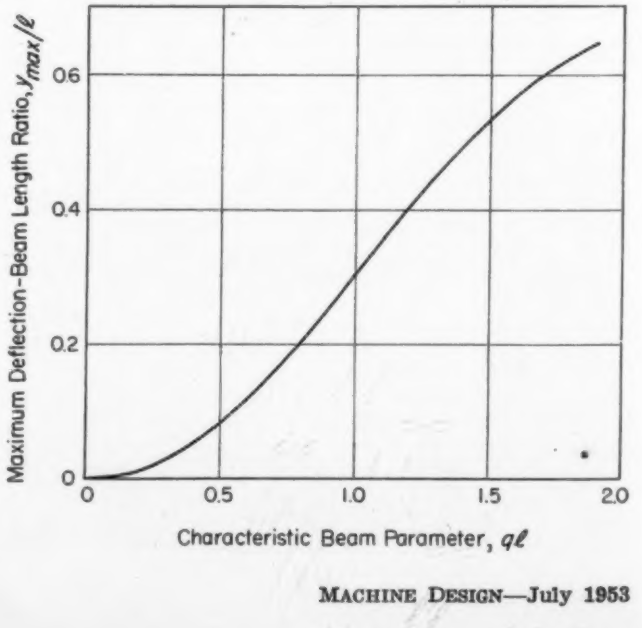


Fig. 3—Maximum deflection as a function of the characteristic beam parameter



are solely in the direction of the applied load. It can be shown that the dimensionless maximum deflection for the elementary theory is given by

$$\frac{\delta_{max}}{l} = \frac{2\tan\beta}{3} \qquad (12)$$

where δ_{max} is the maximum vertical deflection. To compare this result with that of the large deflection theory, Equation 12 can be divided by the similar result y_{max}/l from the large deflection theory obtained from Equation 9. In *Fig.* 4 this ratio, δ_{max}/y_{max}, is plotted as a function of the characteristic beam parameter ql. The values of β corresponding to the various values of ql used for the calculation were obtained from Equation 5.

Since the elementary theory provides no correction for shortening of the moment arm, the maximum stress S_{max} predicted by it may be in considerable error. The magnitude of this error can be determined by calculating the ratio of the maximum stress σ_{max}, as predicted by the large deflection theory in Equation 11, to the maximum stress S_{max}, as predicted by the elementary theory. Equation 11 may be written

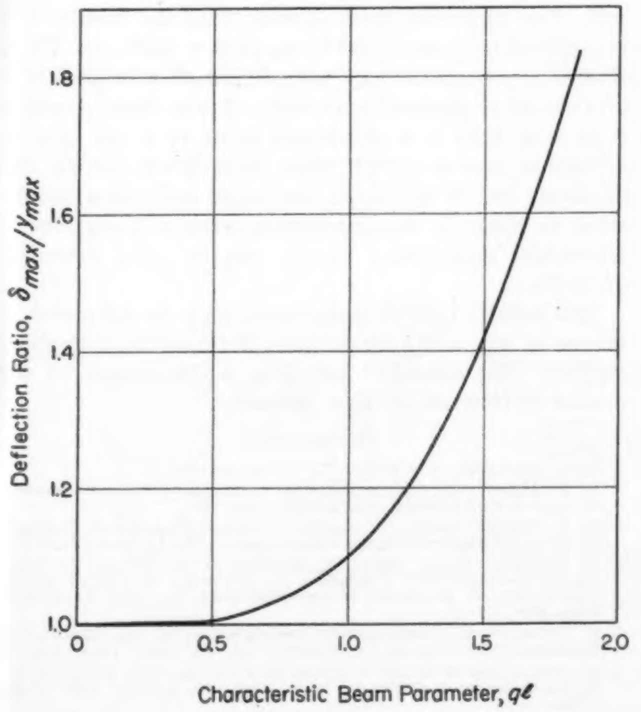
$$\frac{2\,I\,\sigma_{max}}{Flh} = \frac{\sqrt{2\sin\beta}}{q\,l} \qquad (13)$$

Fig. 5 is a plot of Equation 13 utilizing Equation 5 to obtain values of ql for the denominator for corresponding values of β in the numerator.

By the elementary beam theory $S_{max} = Flh/2I$ and this may be shown in the form

$$\frac{2\,I\,S_{max}}{F\,l\,h} = 1 \qquad (14)$$

Fig. 4—Comparison of deflections predicted by elementary beam theory and deflections predicted by large deflection theory



Dividing Equation 13 by this result leads to

$$\frac{\sigma_{max}}{S_{max}} = \frac{\sqrt{2\sin\beta}}{q\,l} \qquad (15)$$

from which it is apparent that

$$\frac{\sigma_{max}}{S_{max}} = \frac{2\,I\,\sigma_{max}}{F\,l\,h} \qquad (16)$$

Thus *Fig.* 5 also represents a plot of the stress ratio σ_{max}/S_{max} as a function of ql.

Design: Results of the foregoing analysis can be readily applied to the design of thin cantilever beams. The following four sample problems and their solutions illustrate conditions that occur frequently in practice.

PROBLEM 1: For a steel cantilever beam $1\frac{7}{32}$ inches long, $\frac{3}{16}$-inch wide, and 0.008-inch thick, find the maximum stress and the deflection at the free end when the applied end-force is 0.2-pound.

The first step is to calculate the value of the characteristic beam parameter ql. This is done by taking the square root of Equation 4 and multiplying both sides by l which results in

$$q\,l = \sqrt{\frac{F\,l^2}{D}} \qquad (17)$$

By Equation 2, since this beam is relatively wide,

Nomenclature

D = Flexural rigidity (Equation 2)
E = Modulus of elasticity, psi
F = Force, lb
I = Area moment of inertia, in.^{4}
M = Bending moment at any point x, lb-in.
S = Elementary beam theory stress, psi
b = Width of thin cantilever beam, in.
h = Thickness of thin cantilever beam, in.
l = Length of thin cantilever beam, in.
$q = \sqrt{F/D}$
ql = Characteristic beam parameter
x = Distance along x axis, in.
y = Large deflection theory deflection along y axis, in.
β = Angle at the end of deflected beam made between the neutral axes of deflected and undeflected positions, deg
δ = Elementary beam theory deflection along y axis, in.
θ = Angle at any point x on deflected beam made by a line parallel to the x axis and a line tangent at x, deg
λ = Distance from clamped end of the beam to the end force F, in.
ν = Poisson's ratio
ρ = Radius of curvature at any point x on deflected beam, in.
σ = Large deflection theory stress, psi

$$ql = \sqrt{\frac{(1-\nu^2)\,Fl^2}{EI}} = \sqrt{\frac{12\,(1-\nu^2)Fl^2}{E\,b\,h^3}} \quad \text{........} \quad (18)$$

where b denotes the width and h the thickness of the beam. Since $E = 30 \times 10^6$ psi for steel and $\nu = 0.3$

$$ql = 0.603 \times 10^{-3}\sqrt{\frac{F\,l^2}{b\,h^3}} \quad \text{....................} \quad (19)$$

Since $F = 0.2$-pound, $l = 1\frac{7}{32}$ inches, $b = \frac{3}{16}$-inch and $h = 0.008$-inch, $ql = 1.06$. From *Figs.* 4 and 5 it is seen that if elementary beam theory is used, the deflections will be more than 10 per cent in error and the resulting stresses will be about 7 per cent in error. It is apparent, therefore, that the large deflection theory should be used. From *Fig.* 5, corresponding to $ql = 1.06$, $\sigma_{max}/S_{max} = 2I\sigma_{max}/Flh = 0.93$ and therefore $\sigma_{max} = 5.58\ Fl/bh^2 = 113{,}000$ psi. Provided the material remains elastic, the deflection at the free end can be found from *Fig.* 3 for $ql = 1.06$ which shows $y_{max}/l = 0.32$ and $y_{max} = 0.39$-inch.

PROBLEM 2: Determine the deflection of a point ½-inch from the free end of the beam under the condition set forth in Problem 1.

For the point under consideration $x = 1\frac{7}{32} - \frac{1}{2}$ and therefore $x/l = 0.59$. From *Fig.* 2, corresponding to $ql = 1.06$ and $x/l = 0.59$, $y/l = 0.16$. So the deflection of the point in question will be l times 0.16 or about 0.2-inch.

PROBLEM 3: Determine the end force that must be applied to the beam of Problem 1 to produce an end deflection of 0.10-inch.

Here a cut-and-try procedure must be used. A value for F will be assumed, and then ql can be calculated by using Equation 19. From *Fig.* 3 the resulting end deflection can be found. If the result is 0.10-inch, the original assumption was correct; if not, the procedure must be repeated until an end deflection of 0.10-inch is obtained.

Assuming that $F = 0.10$-pound, from Equation 19, $ql = 0.75$. $y_{max}/l = 0.18$, so $y_{max} = 0.22$-inch.

Assuming that $F = 0.05$-pound, the foregoing procedure yields $y_{max} = 0.12$-inch.

Assuming that $F = 0.04$-pound, $y_{max} = 0.098$-inch. Thus an end force of about 0.04-pound will produce a deflection of almost 0.10-inch.

PROBLEM 4: Determine the thickness of a steel beam such that the stress in the beam will not exceed the yield stress of 100,000 psi when the applied end force is 0.5-pound. The other dimensions are the same as in Problem 1.

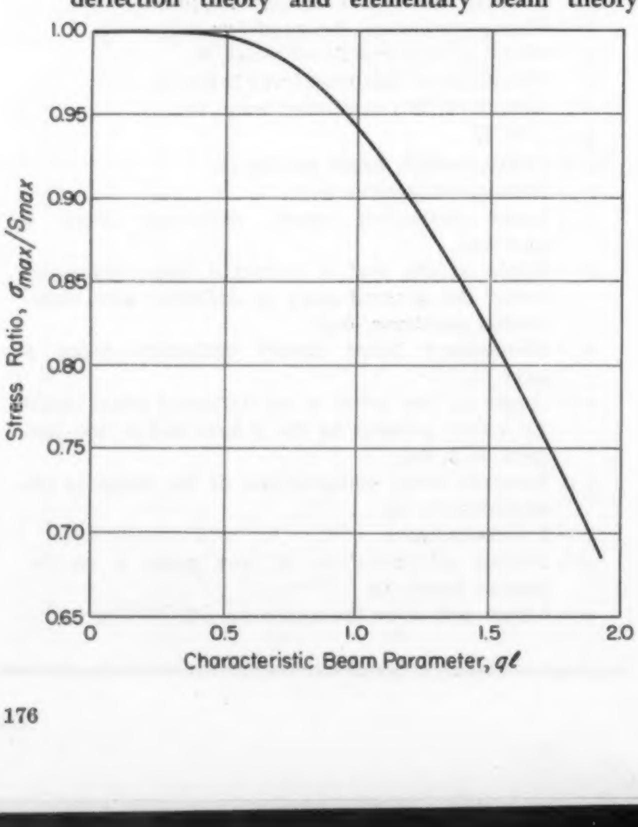
A cut-and-try method must be used here also. Assuming $h = 0.020$-inch and since $\sigma_{max} = 100{,}000$ psi, $2I\sigma_{max}/Flh = 2.05$. From *Fig.* 5 note that the dimensionless stress ratio cannot exceed unity; therefore the assumed value for h is too large.

Assuming $h = 0.012$-inch $2I\sigma_{max}/Flh = 0.74$. From *Fig.* 5, corresponding to this value of the dimensionless stress, $ql = 1.75$. Substituting this value of ql in Equation 19 yields $h = 0.0078$-inch. Since this is not the value of h originally assumed, the procedure must be repeated with a new assumption.

Assuming $h = 0.0139$-inch, $2I\sigma_{max}/Flh = 0.987$. From *Fig.* 5 $ql = 0.66$ and $h = 0.0149$-inch by substitution in Equation 19.

Assuming $h = 0.0138$-inch, $2I\sigma_{max}/Flh = 0.976$. From *Fig.* 5 $ql = 0.79$ and from Equation 19 $h = 0.0132$. The actual beam thickness is between 0.0138 and 0.014-inch which for all practical purposes is 0.014-inch. *Figs.* 4 and 5 show that elementary beam theory could have been used for this problem provided 5 per cent inaccuracy was acceptable.

Fig. 5—Maximum-stress comparison between large deflection theory and elementary beam theory



Conclusions: A solution has been presented for the deflected shapes, the deflections, and the maximum stresses in a thin, relatively wide cantilever beam with a concentrated force applied at the free end. The approach used is such that the results can be applied to a cantilever beam of any stiffness. Thus, elementary beam theory is included as a special case of this more general approach. From *Figs.* 4 and 5 it is seen that if a maximum error of 1 per cent is allowable and $ql < 0.50$, then elementary theory can be used; but if $ql > 0.50$, the large deflection theory must be applied. If a maximum error of 5 per cent is allowable, elementary theory can be used provided $ql < 0.84$.

The results of this data sheet may be extended to beams of any width by suitably defining their flexural rigidity. Timoshenko[5] provides a discussion of the proper definition of this property.

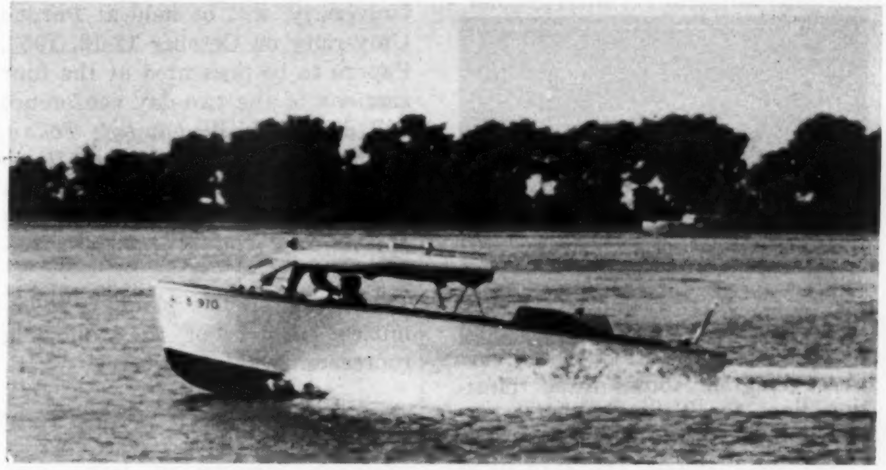
REFERENCES

1. Gross and Lehr—"Die Federn," VDI Verlag, 1938
2. H. J. Barten—"On the Deflection of a Cantilever Beam," *Quarterly of Applied Mathematics*, Vol. 2, 1944, Page 168.
3. K. E. Bisshopp and D. C. Drucker—"Large Deflections of Cantilever Beams," *Quarterly of Applied Mathematics*, Vol. 3, 1945, Page 272.
4. B. R. Seth—"Some Recent Applications of the Theory of Finite Elastic Deformation," *Procedings of Symposia in Applied Mathematics*, Vol. 3; *Elasticity*, McGraw-Hill Book Co., New York, 1950, Page 67.
5. S. Timoshenko—"Mathematical Determination of the Modulus of Elasticity," *Mechanical Engineering*, Vol. 45, 1923, Page 259.
6. E. Jahnke and F. Emde—"Tables of Functions with Formulae and Curves," *Dover Publications*, New York, 4th edition, 1945, Pages 52-85.

Engineering News Roundup

Jet Power Unit For Marine Propulsion

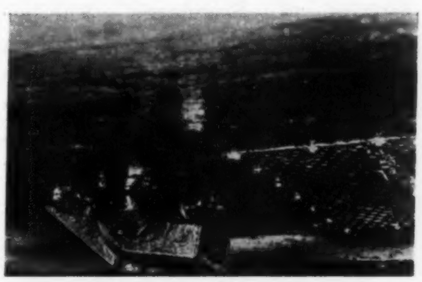
Designed to propel boats from 17 to 21 feet long, a unit consisting of an engine driven centrifugal pump and a swiveling outlet pipe eliminates the need of both propeller and rudder. Water forced through the outlet pipe by the pump drives the boat by reaction or jet propulsion. Labeled the Hanley-Kermath Hydrojet, it is said to increase maneuverability, reduce maintenance and permit operation in shallow and debris-laden water. Efficiency is claimed to be equal to that of a propeller drive when the hull is lightly loaded and better than that of the propeller drive in a heavily loaded hull.



Cruiser driven by Hydrojet propulsion unit powered by a 131-hp engine travelling at high speed

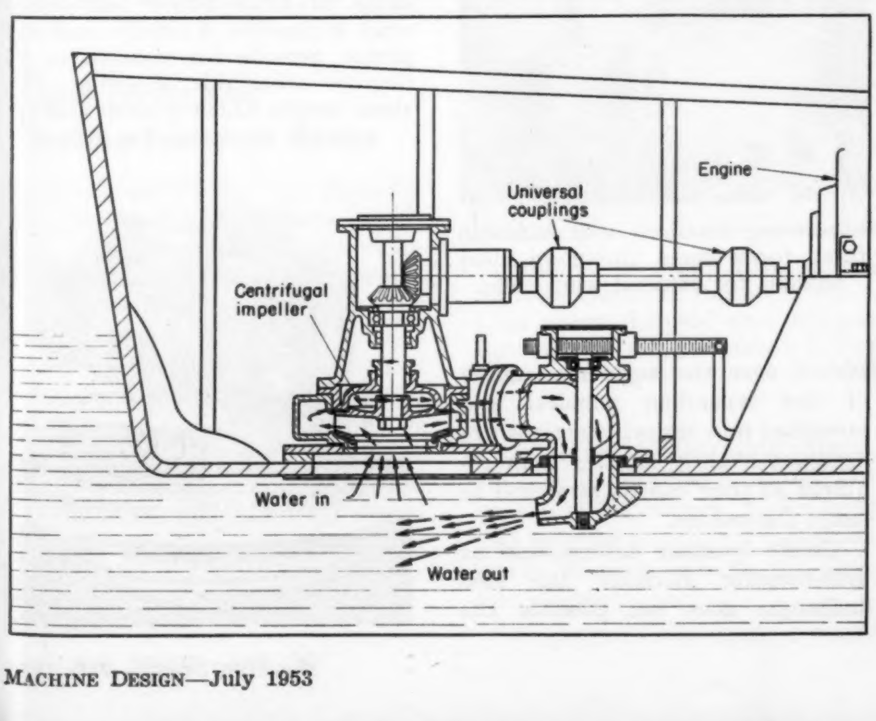
Steering with the Hydrojet is done by rotating the water outlet pipe to change the direction of thrust. Rotation of the outlet pipe also permits reversing without reverse gearing, and power braking. The outlet pipe, or jet nozzle, projects only 3½ inches below the bottom of the hull, making operation in extremely shallow water possible. If the boat should run aground in shallow water, it is said the water jet can be used to wash away the mud or sand and free the craft in a matter of seconds. An additional interesting feature of the unit is that a connection permits pump output to be tapped and used as a high or low pressure stream for fire fighting, cleaning and siphoning bilge water.

The first commercially available units will be powered by a 61-hp Kermath Sea Jeep engine. Larger models are expected to follow.



Underside of hull shows water discharge pipe and inlet screen

Section through hull showing installation of Hydrojet propulsion unit

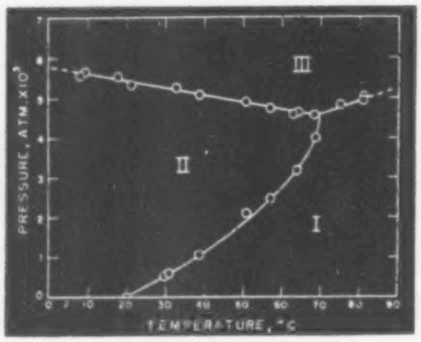


Research Discloses Transitions in Teflon

Teflon (polytetrafluoroethylene) has been found to have three polymorphic forms within the range of 10 to 80 C at pressures between 1 and 10,000 atmospheres. Two different transitions between phases may occur at room temperature depending on the pressure to which the material is subjected. Because these transitions are always accompanied by sharp changes in vol-

ume, the designer should give careful consideration to this property of Teflon when dimensional stability is required.

The first of these transitions occurs at about 20 C or room temperature at atmospheric pressure and is accompanied by a 1 per cent change in volume. The other room



Phase diagram shows three transition lines for Teflon which separate the chart into three pressure-temperature regions. Teflon exists in a different polymorphic form, designated I, II or III in each region

temperature transition takes place at 25.5 C at 5500 atmospheres pressure and involves a volume change of about 2.5 per cent.

This information is the result of research conducted by the National Bureau of Standards. Further technical details are contained in NBS Research Paper RP2395, obtainable for five cents from: Superintendent of Documents, U. S. Government Printing Office, Washington 25, D. C.

Purdue and MACHINE DESIGN Sponsor Mechanisms Conference

To promote better understanding, design and application of mechanisms in machines among draftsmen, designers and other engineering personnel, the First Conference on Mechanisms, sponsored by MACHINE DESIGN and Purdue University, will be held at Purdue University on October 12-13, 1953. Papers to be presented at the four sessions of the two-day conference will deal with *Mechanisms Today, Mechanism Problems in High-Speed Design, Procedures in Mechanism Design* and *Applications of Mechanisms in Automatic Machinery.*

The conference planning committee consists of the following representatives of industry: W. R. Spiller, vice president of engineering, Harris-Seybold Co., W. A. Witham, executive engineer, Miehle Printing Press and Mfg. Co., M. S. Curtis, vice president of engineering, Warner and Swasey Co., G. B.Carson, secretary, Selby Shoe Co., W. A. Patzer, president and chief engineer, ABT Mfg. Co. and R. G. Zuefle, chief engineer, American Machine and Foundry Co. Professors E. S. Ault and A. S. Hall, Purdue University, with B. L. Hummel and R. W. Bolz, associate editors of MACHINE DESIGN, complete the committee.

Complete details of the program including procedure for advance registration will appear in the August issue of MACHINE DESIGN.

Expanding Mandrels Form Large Contoured Components

A new expand forming technique for fabricating aluminum and stainless - steel sheet - metal parts has been developed by Ryan Aeronautical Co. Originally employed for production of large external wing fuel tanks, the method is being extended to the manufacture of large components for General Electric jet engines and other turbine power plants. The new forming process is accomplished by the use of huge expanding mandrels on the largest machine tool of its type. The machine is 17½ feet tall, weighs 34,000 pounds, and exerts a force of 4800 tons.

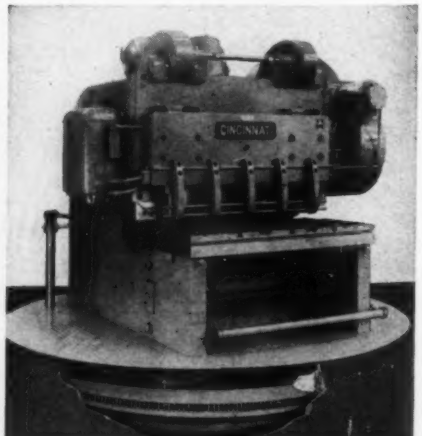
Ryan's new technique permits precision forming of large contoured closed sections to exact dimensions which meet requirements for smooth contours and close tolerances important in supersonic jet-propelled aircraft. Contoured components are fabricated by rolling the sheets into cone shapes, which permits maximum forming with least elongation, and welding these cones automatically. Fusion welds join the metal edges in a single thickness bond of uninterrupted metal. Sections are then

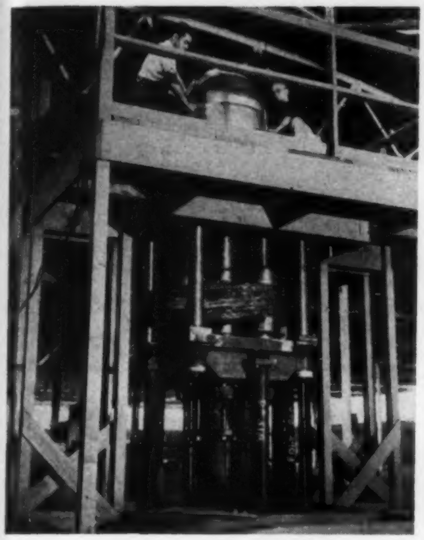
Expanding mandrel with 1200-ton force forms huge aluminum alloy sections for external wing tanks

placed over the segmented shoes of the expanding mandrel and stretched into shape. Sections are designed slightly smaller than required so they can be stretched to exact dimensions.

Unlike hammer deformation or spin-forming methods, the new technique does not produce the warpage typical of hammer forming, which causes an "oil can" effect, and it avoids thinning and consequent weakening of the metal as sometimes occurs in spinning bell-shaped sections. Economy is also a feature, since the aluminum shoes used for forming to different contours are less expensive than other types of tooling, and no ex-

ROTATING SHEAR: Capable of cutting mild steel up to 1 inch thick and 4 feet long, this Cincinnati Shaper Co. shear is mounted on a turntable to eliminate as much handling of large plates and coiled stock as possible. A circular rack and pinion provide for rotation to 90 degrees either side of center. The shear weighs 42,000 pounds and operates at 30 strokes per minute





Largest expanding mandrel, showing entire machine, nine-tenths of which will be below floor level when permanently installed

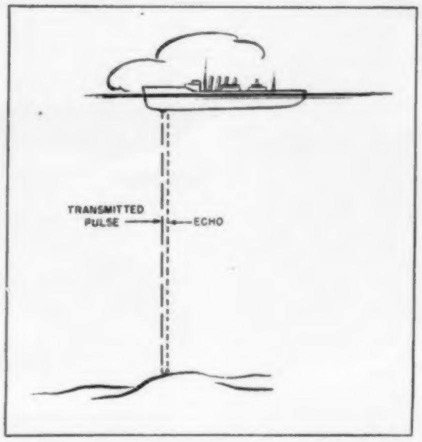
cess material is wasted as flash to be trimmed away.

Source of the mandrel's power is a huge hydraulic ram. Forming the central assembly, an 18½-foot-diameter hydraulic cylinder contains the large piston. Hydraulic fluid under 5000 psi pressure is forced into this cylinder. As it descends, the piston pulls a heavy shaft connected to a tapered pin at the top of the machine. This pin expands a set of eight large nickel-iron segments with aluminum alloy shoes which fit snugly around its circumference. The cone shaped jet engine components are placed over these shoes where the radial force of 4800 tons stretches them into size and shape.

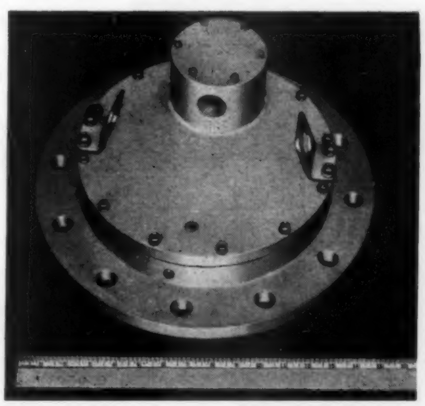
Nickel Tubing Aids Sounding Equipment Design

Magnetostriction, a property which pure nickel possesses, is utilized in recently developed echo-sounding equipment for transmission and reception of supersonic waves from a ship's hull to and from ocean bottom or submerged object. This RCA equipment consists essentially of a transducer, or projector, which is encased in a heavy iron housing and is connected alternately to an energy source and indicating or recording instruments. The transducer sends a pulse of energy in the form of supersonic waves to the ocean bottom and receives the echo of the pulse when it bounces back to the hull of the ship, elapsed time between transmission and reception of the waves indicating depth at that point.

The transducer's pulse of energy is originally an electrical signal, which is converted into supersonic



waves by the vibration of a metal diaphragm. In earlier equipment, the diaphragm was vibrated by means of piezo-electric crystals, which change dimension when electrically charged and generate electricity when pressed or squeezed. The more rugged magnetically polarized nickel, when placed within

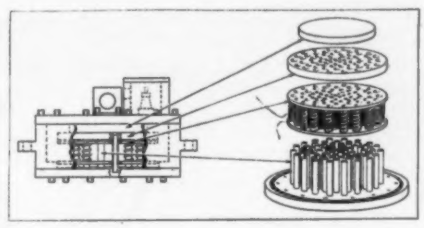


Transducer in heavy iron housing converts electrical energy to supersonic waves. Elapsed time required for waves to travel to ocean bottom and return is proportional to depth

a coil carrying alternating current, vibrates longitudinally at the frequency of the current. Nickel could also be used to convert supersonic waves echoing back to the ships into electrical energy.

Nickel supplied by Superior Tube Co. in the form of seamless tubing reduces the amount needed of this somewhat expensive material and effects savings in weight. Also, slots are cut longitudinally in each tube throughout most of its length, reducing magnetic flux loss. The tubing is cold drawn to ⅜-inch OD by 0.020-inch wall thickness, cut to length and soldered into countersunk holes in the diaphragm plate. Length of the tubing is one-fourth the wave length of the alternating current signal, this length giving a very large amplitude of vibration with a small amount of electrical energy.

In the equipment illustrated, the plate contains 70 lengths of nickel

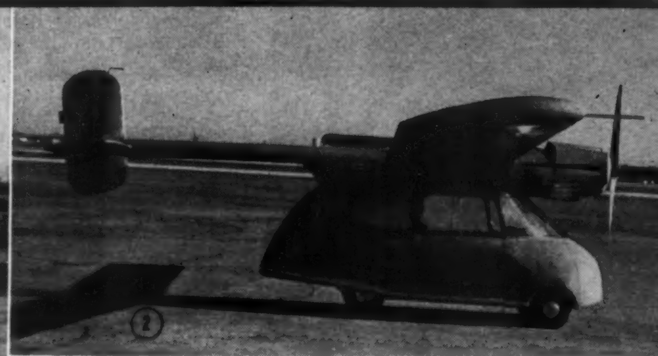


tubing, each of which is surrounded by a coil. When the equipment is in operation, the tubes vibrate simultaneously to produce a piston effect on the diaphragm, creating supersonic waves which measure the depth of water.

Small Atomic Pile Provides Research Medium

Measuring only 5 feet on a side, a small atomic pile was recently placed in operation as a research tool at the Knolls Atomic Power Laboratory operated by General Electric Co. for the Atomic Energy Commission. Although this low-energy reactor cannot be used for making plutonium or for the production of atomic power, it is a useful source of neutrons which can be utilized in many fields of fundamental research.

A unique feature of the reactor is an internal "thermal column," where the neutron flux is much greater than in the fuel region surrounding it. Fundamental proper-



EXPERIMENTAL AIRCRAFT, PAST AND PRESENT: Both the orthodox and unique are present in this array of aircraft first flown at the Experimental Flight department of Consolidated Vultee Aircraft Corp. at San Diego. The flying auto (1), air car (2), Stinson Sky Coach (3) and controllable wing craft (4) appear to be attempts to make an airplane "roadable" (or an automobile "flyable"). More recent models are the XF-92 (5), world's first delta wing plane, the XP5-1 turboprop seaplane (6), the XB-46 jet-powered bomber (7) and the turboprop powered Turbo-Liner (8)

ties of the various kinds of atoms which are placed in this column can be determined quickly and easily. Thus the reactor can detect minute amounts of many elements—quantities considerably below the capabilities of ordinary methods of analytical chemistry. At comparably low reactor power levels, it can also make small quantities of radioactive forms of elements that are ordinarily nonradioactive.

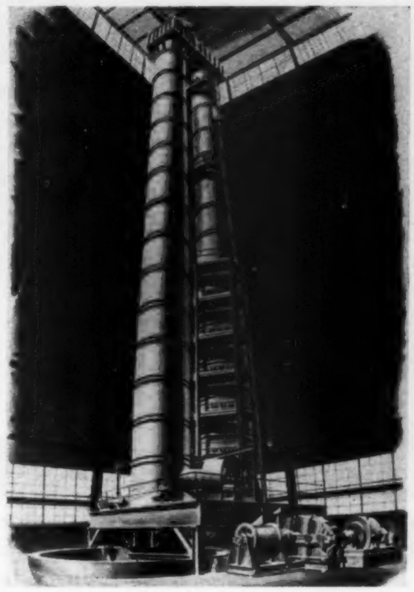
The new thermal test reactor, which is inside a cell surrounded by a 6-foot-thick concrete wall, consists fundamentally of a cube made of blocks of graphite, with the atomic fuel in an 18-inch aluminum tank inside a cylindrical hole in the center of the graphite. Fuel is "enriched" uranium, in which the amount of fissionable U235 has been increased. Only about 6 pounds of such fuel are required.

In operation, splitting of the U235 atomic nuclei releases neutrons at high speed. Passing through the graphite, which serves as a "moderator," the neutrons are slowed to "thermal" speeds of about a mile per second, the same as those of oxygen and nitrogen atoms in the air. Some of these thermal neutrons, hitting the uranium fuel elements, cause fission of more U235 nuclei, thus maintaining the chain reaction.

Fuel is contained in 2-inch disks of an alloy of uranium and aluminum, strung on rods and kept apart by plastic separators. Each rod, which is about 18 inches long, is hung in a can containing a light paraffin base oil. Twenty such cans are arranged in a circle, with water between them.

In the middle of this ring of fuel cans is a graphite cylinder 12 inches in diameter, with a test hole in the center. Materials to be tested or exposed to neutrons are placed in this hole and subjected to a strong neutron bombardment as the reactor is operated.

Control of the reactor is accomplished with rods and sheets of cadmium which can be lowered into the fuel assembly, absorbing neutrons and so preventing them from carrying on the chain reactions. Other safety rods drop into place automatically when needed.



Engineering News

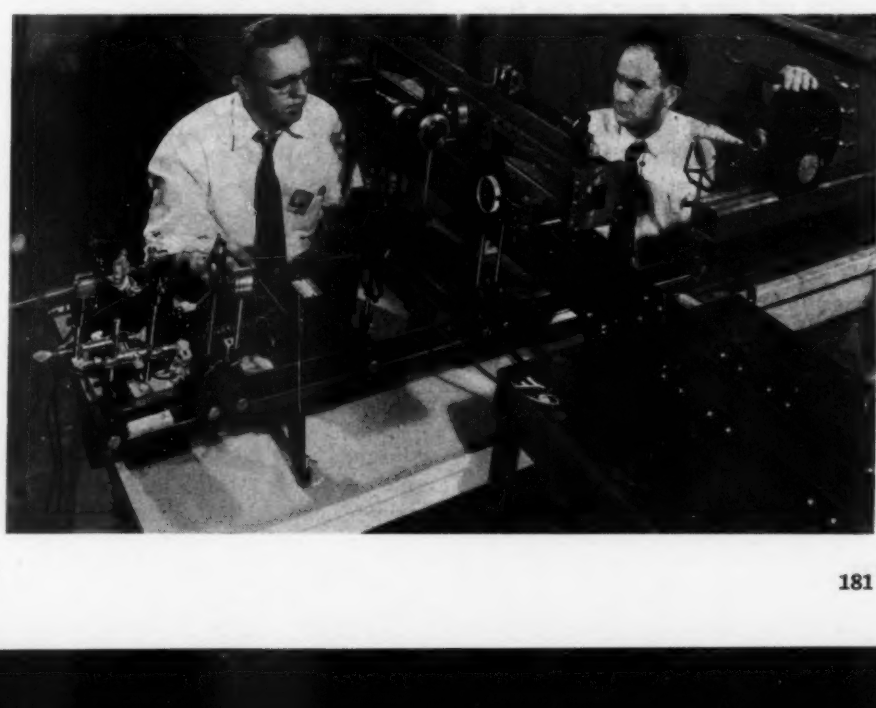
13 - STORY - HIGH FURNACE: Artist's drawing of a furnace being built by Westinghouse which will have a 130-foot vertical tower and will be used to heat-treat long extruded aluminum shapes used primarily in the manufacture of aircraft. Pieces 110 feet long will be suspended in the tower for specified periods of time, and at the end of the heat bath, will be quenched in a water-filled pit of the same depth. Temperature difference between top and bottom of the tower will be held to a maximum of 5 degrees by maintaining a high rate of air circulation. The furnace is scheduled for completion in about a year

Sintered Nylon Bearings Operate Without Lubrication

Sintered nylon bearings in a recent test withstood 18 months of continuous use without any visible evidence of wear and without any lubrication except for the few "break-in" drops of oil applied at the time of installation. Six of the new type bearings, first announced less than a year ago, were installed in place of bronze bearings on the conveyor belt roll feed and horizontal backshaft of a machine which has been in operation 40 hours a week since installation.

All six bearings were made from Nylasint molding powder, Type 66, a finely divided nylon powder supplied by National Polymer Products Inc. Four bearings used on the conveyor belt roll feed are ½-inch ID, ⅝-inch OD and ⅝-inch long, with a ¾-inch OD flange at

JET COMBUSTION STUDY: Miniature shock waves similar to those created by an atomic blast aid engineers in the study of jet engine combustion processes. The shock waves are produced by General Electric engineers in a shock tube composed of two chambers which have glass windows for high-speed photographic observation. Pressure is built up in one chamber and released in the other. A thin diaphragm separates the chambers, and when this diaphragm is punctured, the difference in pressure sends a shock wave the length of the tube at supersonic speed. The camera records the shock wave for future study of the effect of the waves on combustion processes



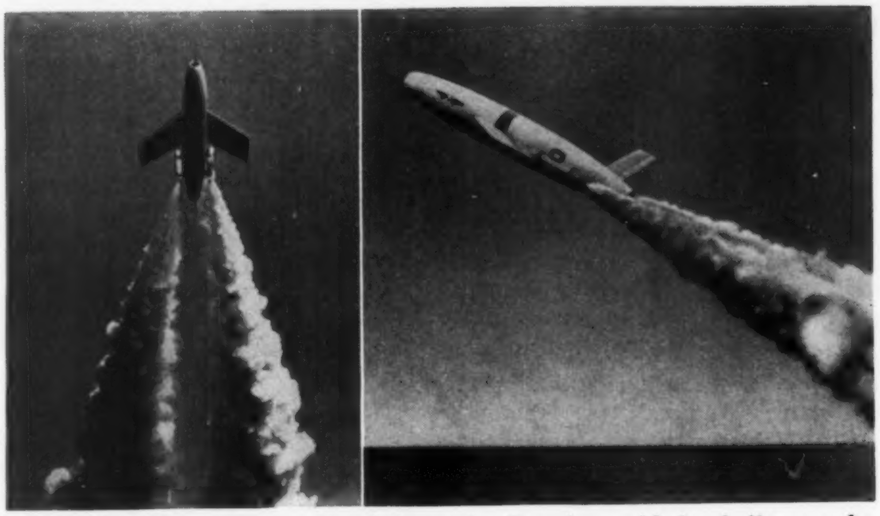
one end. Each one had a 0.001-inch press-fit in a steel housing. The roll feed operates at 460 rpm under a light load with 0.003-inch clearance. Two stationary bearings used on the horizontal backshaft measure 5/16-inch ID, ½-inch OD and ⅞-inch long, and are press-fitted approximately 0.001-inch into small steel blocks. The shaft operates at 250 rpm under a very light load with 0.0015-inch clearance.

The powder used in making the nylon bearings is produced by a chemical process that makes the material suitable for cold pressing and sintering. It lends itself readily to the production of sintered nylon bearings, gears, cam rollers, valve seats and other industrial products. Laboratory and field tests indicate that bearings made by cold pressing and sintering nylon powder have an unusually low coefficient of friction and show promise for use in frictional applications where operation without lubrication is required.

Competition Returns To World Steel Market

The world market for steel is now going through a change and is becoming competitive again, according to Michael J. Layton, head of the International Relations Department of the British Iron and Steel Federation. In a speech before the general meeting of the American Iron and Steel Institute Mr. Layton declared that the production of steel in western Europe, which reached a rate of over 71 million net tons a year in the first quarter of this year, appears to be overtaking demand.

In the United States, as well as in Europe, new plants should soon raise production to a level which will meet domestic demand and make possible re-entry into the export market. In Europe, capacity this year and next should be well ahead of requirements. Raw materials, although unable to keep full capacity going, will probably be more than sufficient. The market is becoming competitive again, and prices, cartels and other market arrangements are reappearing, says Mr. Layton.



Resembling a swept-wing jet fighter, the Regulus guided missile may be launched from sub, ship or shore. Jato units are used to provide additional thrust for launching

New Guided Missile Launched by Sub

The existence of a guided missile which may be launched from a submarine as well as from surface ships or shore installations was recently announced by the Department of Defense. Named the Regulus after a star of the first magnitude in the constellation Leo, the missile was developed by Chance Vought Aircraft Div. of United Aircraft Corp. under the sponsorship of the Navy Bureau of Aeronautics. In appearance, Regulus resembles a swept-wing jet fighter. The 30-foot-long missile has completed operational tests and is now in production, Performance data have not been released; however, performance is said to exceed design specifications.

A World War II submarine, the Tunny, has been modified to launch the Regulus. In addition to the installation of a tank for stowing the missile and a launching rack, the Tunny has been modernized by adding a snorkel and streamlining the hull and conning tower.

Miniature Device Controls Jet Engines

Similar to a miniature jet engine, a new instrument makes the operation of high-speed jet aircraft easier and safer. Replacing complicated electronic equipment, the Solar Aircraft Co. Microjet control directly senses engine pressures and quickly compensates for variations in pressure.

When in action on a jet engine, the pneumatically operated device computes the correct turbine discharge pressure under all flight conditions and at the same time notes any error between the actual engine pressure and correct engine pressure. If there is an error, the Microjet automatically sends out electrical signals to other engine controls which correct the pressure conditions. Pressure variations are recognized in 0.01-second.

To be used by Westinghouse on J46 jet engines and by other companies for testing and experimental

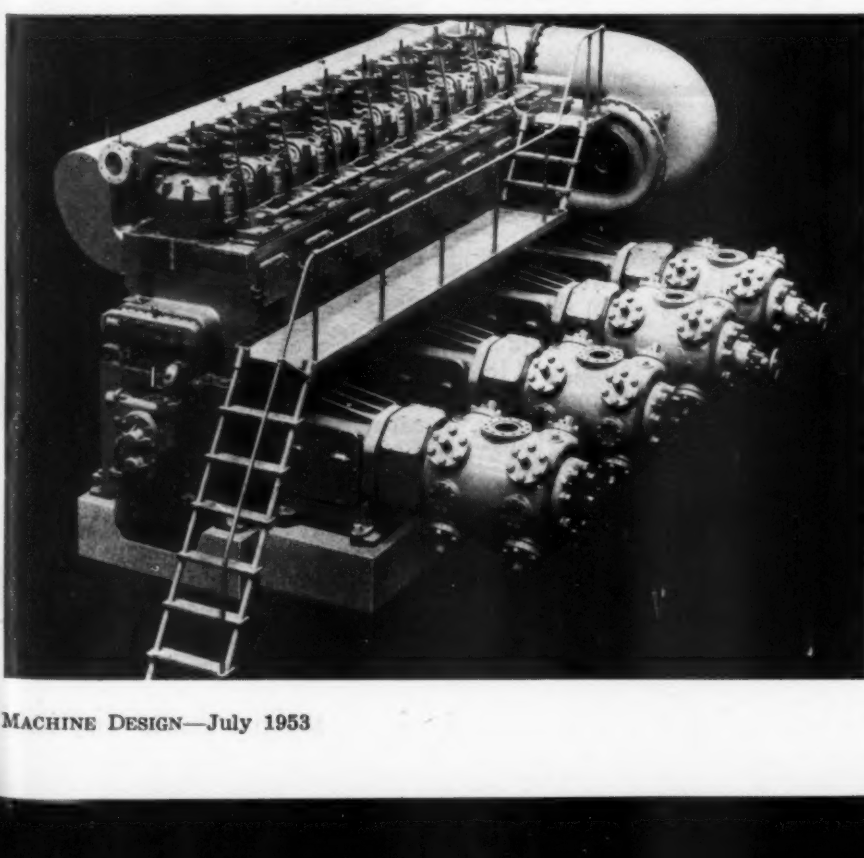
purposes, the new control is claimed to have important applications on modern aircraft engines. Various industrial uses are also being explored.

MACHINE DESIGN Wins Top Editorial Award

Known as the "highest award in business publication journalism," the bronze plaque of the *Industrial Marketing* editorial achievement competition has been won by MACHINE DESIGN for the October, 1952 issue on Engineering Materials, which was judged the year's "best single issue." In the same competition MACHINE DESIGN also won three second-place certificates of merit, one for Dorian Shainin's recent articles on "Quality Control" (best series), one for the 1952 front covers (best graphic presentation), and one for Jack Greve's April, 1952, article on "Electric Motor Drives" (best original research). The four awards were presented to Colin Carmichael, Editor, during the convention of the National Industrial Advertisers Association in Pittsburgh, June 22-25.

In winning four awards in a field of 515 entries, MACHINE DESIGN is the only design publication to be so honored this year. Last year MACHINE DESIGN was awarded two bronze first-place plaques in this same competition, one for the 1951 April issue (best single issue) and one for Randolph Chaffee's article on "Evaluating Engineers" (best single article).

TURBOCHARGED COMPRESSOR: A unique two-cycle, right-angle, gas-engine driven unit boasts high thermal efficiency and economy of operation. Designated Model TRA by Clark Bros. Co., it produces 50 per cent more power than nonturbocharged units of comparable bore and stroke and requires 25 per cent less cooling water per bhp. Turbocharging converts waste heat, velocity and mass flow of exhaust gases to driving force for a radial inflow turbine. The turbine is integrally connected to a centrifugal compressor which pumps air for scavenging the power cylinders and also provides air for combustion



New American Standard To Help Control Noise

Claims totalling $2 billion for deafness due to excessive industrial noise are in the courts today. Noisy machines are creating accidents because warnings and instructions are not heard or are misunderstood. Sales resistance to noisy machines is naturally developing.

The first step was taken recently in solving the most urgent noise problem — measurement of the noise. As a result of a research program, a committee of the American Standards Association has determined that measurement of overall noise level is not adequate, since damaging effects also vary with frequency. Low frequency noise has much less harmful effect than high frequency noise of equal intensity. To promote uniformity of analysis, a new American Standard Specification, Z24.10-1953 for an Octave Band Filter Set for the Analysis of Noise and Other Sounds, has been prepared. Such an analyzer divides the audible spectrum into eight bands.

A subcommittee of ASA committee Z24 is presently analyzing available information to derive criteria with which to compare sound level measurements in order to determine whether or not sound level is excessive. If this comparison shows the need for a reduction in noise, changes in a noisy machine can then be made to accomplish the reduction.

Alloy Steel Develops New Strength

Heat treating and processing procedures recently perfected by Lockheed engineers make possible the effective use of standard, deep-hardening 4340 steel with a tensile range of 260,000 to 280,000 psi. Formerly utilized at 180,000 to 200,000 psi, 4340 steel achieves maximum efficiency at the new, higher strength level.

Such vital aircraft parts as cargo and passenger plane landing gear are now being made stronger and lighter through use of the 40

per cent higher tensile strength. The increase in the metal's strength, coupled with a 23 per cent weight saving over the conventional heat-treated steel in comparable cross-section structures, makes possible greater payloads for commercial transports.

First phase of a standard heat-treating cycle adopted for all parts includes normalizing of the material at approximately 1600 F to produce a homogeneous structure with complete solution of carbides.

Austenitizing is accomplished by heating the material to 1500 F, a temperature which keeps the retained austenite to a minimum upon quenching in oil. Parts ranging from more than ¼-inch to 1 inch diameter are heated for 45 minutes and then held for approximately one hour. Parts from 1 to 2 inches diameter are heated for approximately 1¼ hours and held for 2 hours. Drawing is performed for 4 hours per inch of cross section, with a 4-hour minimum, at temperatures of 400 to 500 F.

As a final thermal operation, all parts are heated at approximately 250 F for 24 hours to stabilize any residual austenite.

Teflon-Faced Rails Protect Conveyed Products

Damage caused by adherence to and abrasion of conventional guide rails of mechanical conveyors can be largely eliminated by the use of new Teflon-faced steel guide rails. A method of cementing the plastic to steel plate is a development of the Fluorocarbon Products Div. of the United States Gasket Co. The Teflon surface of the rails is ⅛-inch thick, and the steel backing plate is ⅛-inch or thicker, depending on requirements. The rails are available in any width and in sections up to 36 inches long.

Teflon, called the slipperiest plastic known to science, is also chemically and biologically inert, tasteless, odorless, noncontaminating and long wearing.

PRINTED CIRCUITS: The glass negative being inspected by an RCA technician will be used to photo-etch electronic components in a new process quite similar to that used to make the plate from which this picture was printed. Copper-clad sheets of plastic are coated with a special emulsion, exposed and etched to give a copper positive. The process will faithfully reproduce circuits having line widths of 0.01-inch and any number of identical circuits may be made from one negative

Aluminum Radiators Predicted for Automobiles

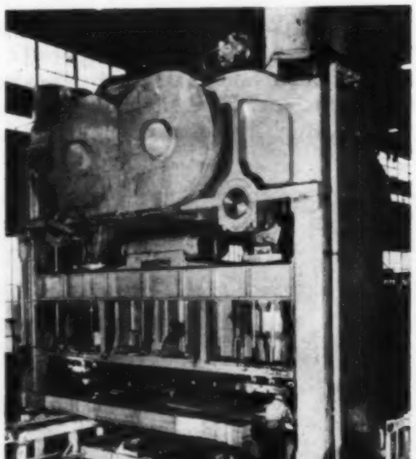
Research by automotive and aluminum manufacturers will lead to greater use of this metal in motor vehicles of all types, according to the Aluminum Association, and the use of aluminum radiators is expected to become economically feasible in the forseeable future. One manufacturer has 35 such radiators in use, all of which have been operating satisfactorily for over a year under severe conditions, with individual mileages ranging up to 30,000 and more.

Much of the research on this use of aluminum has been aimed toward developing the best method of fabrication. A brazed assembly promises to serve satisfactorily, since it can be operated at all cooling system pressures currently being considered, a characteristic which has been in demand by automobile manufacturers. Other possible methods of fabrication are also being investigated, including soldering aluminum fins to aluminum or brass tubes.

Largest Mechanical Press For Auto Chassis Frames

To produce side rails with less camber, or "bow," automobile chassis frame manufacturers seek increasingly heavier presses, which operate with less deflection in the bed and slide. Such a press is in production in the Cleveland plant of Midland Steel Products Co. Built by Baldwin-Lima-Hamilton and weighing 1.8 million pounds, it is reportedly the largest mechanical press ever built.

Design of this press incorporates "negative deflection," which makes possible a reduction of weight in



the bed and slide. When loaded to 4000 tons—the nominal pressure capacity—no relative deflection occurs in the bed and slide. As a result, the formed side rail is perfectly straight.

Construction of the frame is the conventional four pieces, but instead of the tie rods being heated and shrunk in the usual manner, they are cold stressed with hydrau-

JOHNSON MODERNIZED SLEEVE BEARINGS

SOLVE MANY SPECIFIC PROBLEMS

The sleeve bearing is basic in principle . . . for most applications man's genius has never invented any device to replace it. For instance, every internal combustion engine is dependent on it, and the majority of industrial equipment is, too. But modern requirements, high speeds, higher loads, and changing designs have called for new sleeve bearings. Johnson Bronze, alert to the demands of new engineering trends, is producing Johnson Modernized Sleeve Bearings, capable of giving the utmost service under these modern conditions. Aluminum bonded to steel is one of them. Copper-lead bronze-on-steel is another. New materials, new combinations of materials and new designs are meeting many new requirements. If you have a specific sleeve bearing problem, perhaps a Johnson engineer can help you. Write for an appointment.

JOHNSON BRONZE COMPANY
525 South Mill Street, New Castle, Pa.

JOHNSON JB BEARINGS
Sleeve-Type

JOHNSON BRONZE PRODUCES ALL TYPES OF SLEEVE BEARINGS: ALUMINUM-ON-STEEL • CAST ALUMINUM ALLOY • BRONZE-ON-STEEL, copper lead • STEEL BACK, babbitt lined • BRONZE BACK, babbitt lined • CAST BRONZE, plain or graphited • SHEET BRONZE, plain or graphited • LEDALOYL powder metallurgy

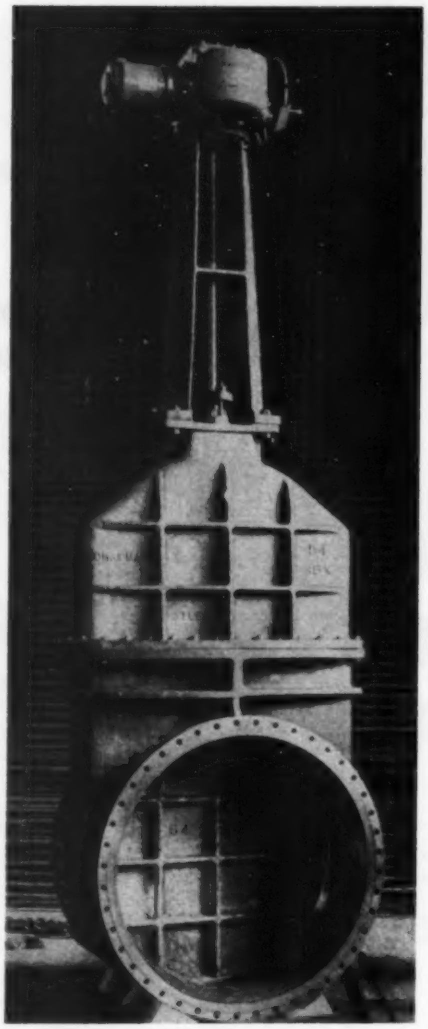
lic jacks. Thus stress in the rods is predetermined, offsetting working stresses in the press.

Bed of the press is equipped with six 42-inch diameter air cushions, providing 415 tons pad pressure with air at 100 psi. Air cushions were selected because their stripping power is always equal to their drawing power. The six cushions are linked together, and the six operating valves are controlled by one pilot valve; therefore the cushions function as a unit at all times.

Jet Trainer Design Permits Engine Interchangeability

Winner of a recent design competition conducted by the Air Force, the Ryan T-59 primary phase jet trainer may be powered by an Allison 520-C1 or two French Marbore 351 jet engines. Also, the design is flexible enough to accommodate any jet power plant in the same power class which may be built in the near future.

Purpose of the new trainer is to provide dual instruction for student pilots making the transition from relatively slow aircraft powered by reciprocating engines to high speed jets. Performance of the T-59 fills the gap between the two. Maximum speed is 427 mph at sea level and 438 mph at 35,000 feet when powered by the 520-C1 engine, or 410 mph at sea level and 428 mph at 35,000 feet when two Marbore engines are used.



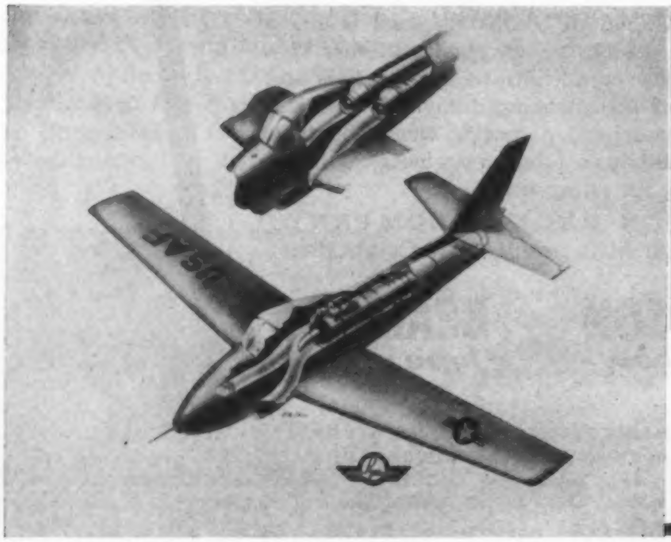
GIANT GATE VALVE: Made entirely of 18-8 stainless steel, this valve is 19 feet high and weighs 9 tons. One of the world's largest, the valve was made recently by the Chapman Valve Mfg. Co. to handle water contaminated with highly corrosive agents.

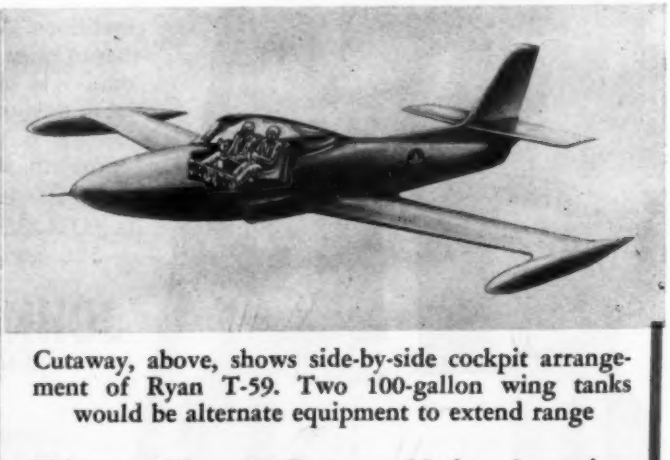
Organize National Fluid Power Association

The National Fluid Power Association was organized recently following a meeting of 45 companies in the fluid power field—consisting essentially of manufacturers of fluid power components, such as hydraulic and pneumatic valves, cylinders and pumps.

Among the stated purposes and objectives of the association are participation in activities which may further the application of fluid power to the machines of industry, co-operation with joint industry committees to advance performance and application of fluid

"We have nothing but the greatest respect for your work, Jones —absolutely nothing."

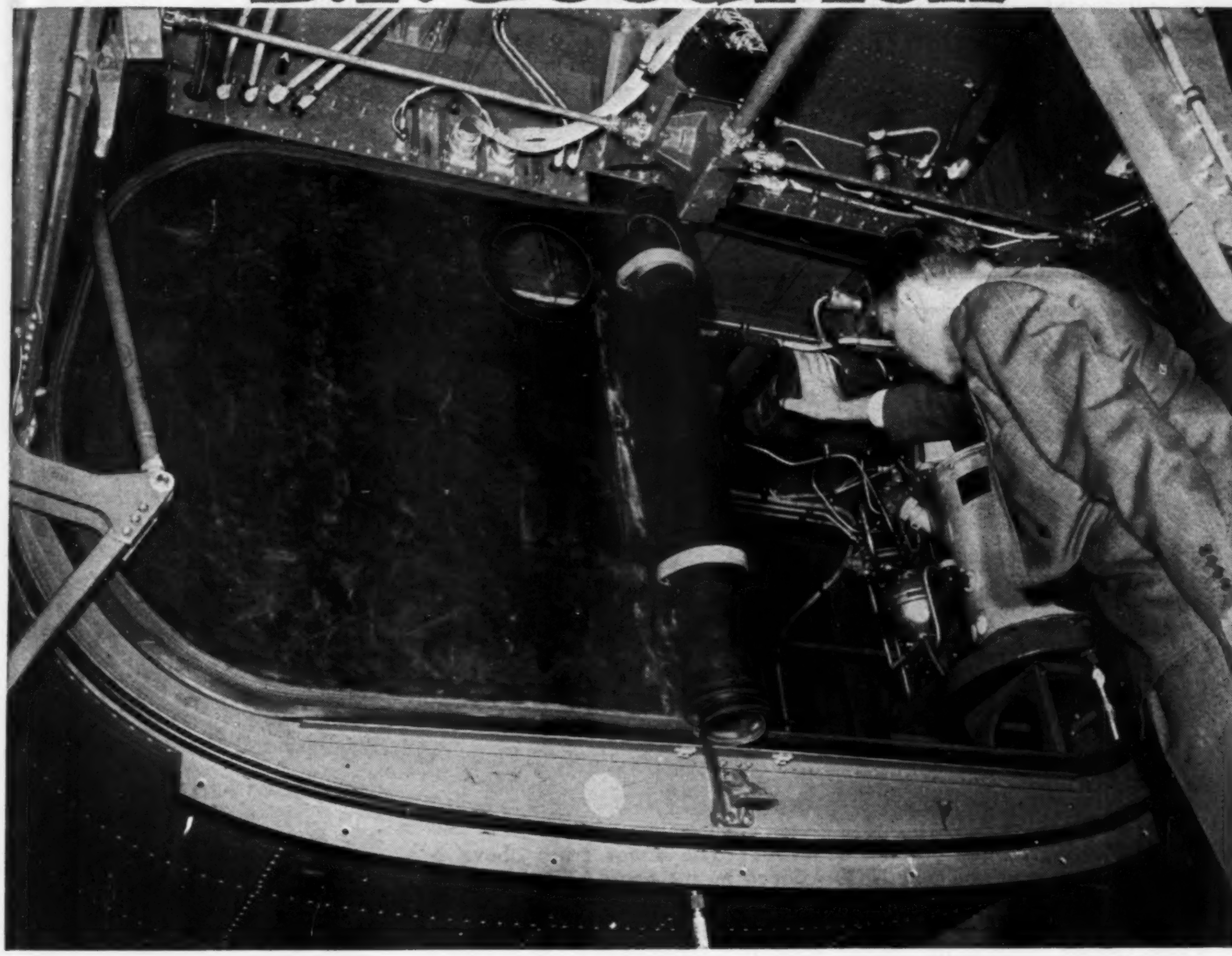




Cutaway, above, shows side-by-side cockpit arrangement of Ryan T-59. Two 100-gallon wing tanks would be alternate equipment to extend range

Either an Allison 520-C1 or two Marbore jet engines may be used to power the Ryan T-59, left

B.F.Goodrich



New zipper seals tight, zips open fast

A NEW *KIND* OF SEAL keeps gases, liquids, dirt *in* or *out*. In the picture, it is used to seal a curtain that keeps fumes from an airplane's engines out of the forward compartment. Getting inside in a hurry was often as important as sealing out the fumes. Other methods of sealing called for screws or other fasteners. They took time to remove. The new B. F. Goodrich Sealing Zipper did both jobs—sealed tight, opened fast.

Here's how it works: Rubber lips are molded with hairline precision so that they fit together tightly. They press together even without pressure to seal effectively. And with pressure against the lips, the seal actually tightens. The rubber seal is applied to an ordinary zipper which zips open or shut in a hurry.

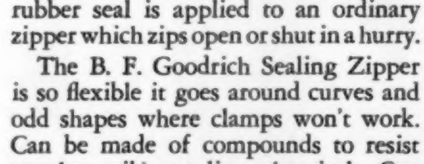
The B. F. Goodrich Sealing Zipper is so flexible it goes around curves and odd shapes where clamps won't work. Can be made of compounds to resist weather, oils, gasoline, chemicals. Can be sewn or cemented to fabric, metal, wood, glass. The rubber can be made to match any color. Production costs are often reduced because the zipper is so easy to install.

It has been used in life-saving suits, to seal delicate instruments against moisture, to seal lifeboats against sea water, to seal auto convertible tops against rain. Write for complete details, sending blueprints or specifications of your products to Dept. A-40 or send coupon below. The B. F. Goodrich Company, Zipper Division, Akron, Ohio.

B.F.Goodrich

Airtight, Watertight Sealing Zipper

OVERLAPPING RUBBER LIPS IN THIS CROSS SECTION SHOW HOW ZIPPER SEALS



Send Now for Free Folder

Gives case histories, design advantages, suggested uses.

The B. F. Goodrich Company
Dept. A-40, Akron 18, Ohio

Name ....

Firm ....

Title ....

Address ....

City .... *Zone* .... *State* ....

power components, and the advancement of knowledge and understanding in the field.

John C. Hanna, vice president of Hanna Engineering Works, was elected chairman of the new association. Headquarters office has been established at 1618 Orrington Ave., Evanston, Ill.

Titanium Resists Galvanic Corrosion

Experiments made to determine the galvanic corrosion behavior of titanium under varying conditions prove that the new metal is highly corrosion resistant when coupled with other metals. Galvanic corrosion tests in which titanium, made in the Bureau of Mines laboratories, was joined to other metals in acid and salt solutions show that titanium does not corrode when coupled with magnesium, zinc, aluminum, iron, copper or stainless steel in a 3 per cent sodium chloride solution—nearly the same as sea water.

Titanium corrodes when coupled to copper in sealed bottles containing various concentrations of hydrochloric acid aerated by helium. When air is substituted for helium, however, the copper corrodes and the titanium is protected. Adding dissolved copper to the solution increases corrosion resistance of the titanium.

Detailed results of the experiments are given in the Report of Investigations 4965, *Certain Aspects of the Galvanic Corrosion Behavior of Titanium,* a free copy of which can be obtained from the Bureau of Mines, Publications Distribution Section, 4800 Forbes St., Pittsburgh 13, Pa.

AUTOMOTIVE AUTOMATION DISPLAY: The motor truck above is actually a display room on wheels. Developed by the Standard Control Division of the Westinghouse Electric Corp. to take displays of control equipment directly to design engineers at the plants where they are employed, it is now visiting plants in the Middle West. It is expected that the 1½ hour demonstration program will be given at 350 plants over a period of 14 months. Operation of a control system is being explained below

Neg'ator Exerts 400-Pound Force

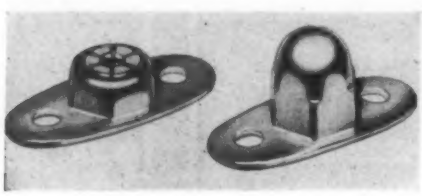
Formed for a military application by the Hunter Spring Co., the largest constant-force spring ever produced exerts a force of 400 pounds throughout a distance of 75 feet. Made of a single section of stainless steel 84 feet long, 10 inches wide and 0.10-inch thick, and formed by prestressing flat spring material so that its unstressed condition is a tight coil, the large neg'ator has an energy storage capacity of 30,000 ft-lb.

A short length of the large neg'ator is shown as a double-coil clamp, on top of which rests the smallest neg'ator yet made, having a force output of 11 ounces which remains constant through a distance of approximately 1 inch.

NEW PARTS

Anchor Nuts

Light in weight and having small envelope dimensions, these two self-locking anchor nuts are made with 4-40 and 6-32 size threads.

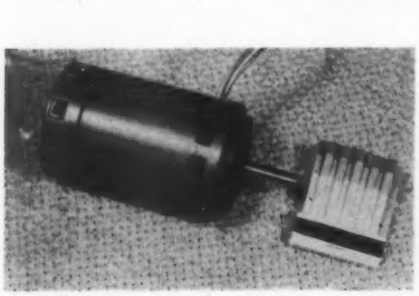


They meet requirements of AN-N-5 and AN-N-10 specifications. Nuts are similar except one has a dome which protects against wire chafing on screw threads in electrical assemblies. Made by **Nutt-Shel Co.**, 811 Airway, Glendale 1, Calif.

For more data circle MD-36, page 191

Small Ac Motors

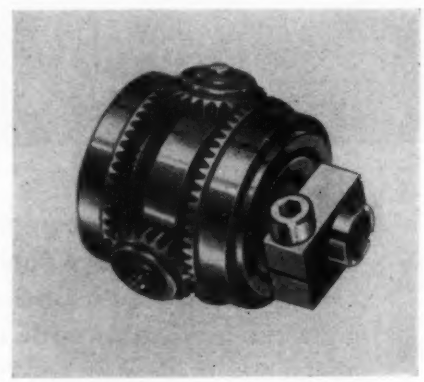
The small motors in this series are precision units for special service applications. The stepped series of 8 lamination sizes allows complete coverage from the smallest instrument service to medium power duty. There is great flexibility within each of the 8 frame diameters. In addition, they may be custom built to meet a manufacturers requirements and specifications. Basic design and construction are particularly applicable for operation on 400 cycles, however designs for operation at 2 to 1600 cycles as well as variable frequency motors are also available. Basic design variables are utilized to attain such types as precision servo motors, lightweight blower motors, pump and antenna motors, etc. Motors may be totally submersible, with a corrosion resistant finish, and can be designed to meet high temperature requirements. To make the motors particularly applicable for manufacturers engaged in defense work, they have been designed to meet exacting military specifications. Made by **Hertner Electric Co.**, 12690 Elmwood Ave., Cleveland 11, O.



For more data circle MD-37, page 191

Hollow Shaft Differential

A hollow shaft differential of low inertia (0.0745 oz-in.), minimum backlash (0.10-in. max) and low weight (1¼ oz) has been developed for use in computers. Designed for high accuracy in additive and subtractive operations, the mechanism will have primary application to angular or angular velocity sums and differences, and sequence operations. The versatility of the hollow shaft design permits the axial positioning of the differential on the shaft. It also facilitates easy installation or removal of the differential from the shaft



without differential or instrument disassembly. Shrink-on, safety-keyed side gears are available to specifications. Built with precision ball bearings the unit has an overall axial length of 1 3/16 in. and receives a 3/16-in. diameter shaft. Made by **Librascope Inc.**, 1607 Flower St., Glendale, Calif.

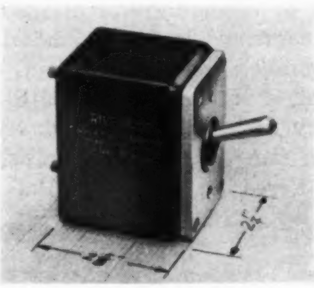
For more data rcle MD-38, page 191

Waterproof Toggle Switch

In addition to being waterproof, the switch is dust-tight, quickly and easily installed, functions from −65 to 165 F, and withstands shock, vibration and salt spray, according to the manufacturer. The switch mechanism is encased in a rubber compound especially

developed and molded. A rubber boot acts as a waterproof seal around the toggle and between the mounting plate and the switch itself. Continuous rating of the switch is 175 amp at 15 or 30 dc or 125 v ac. Maximum inrush capacity on any type of load is 500 amp. Toggle and pole arrangement is SPST.

In one of its ordnance applications the switch is "ganged" in pairs to handle 300 amp. This is



accomplished by using a gang bar attached to both toggles, which actuates them as a single toggle, and through the use of a connecting device between the poles to make them a common pole. The switch can also be wired in multiple as DPST and will be available shortly in a SPDT arrangement as well. Over-all dimensions are 2¼ by 2⅝ by 3½ in. with the toggle extending 1 7/16 in. beyond the mounting plate at the top of its travel. Weight is approximately 1½ lb. Made by **Riverside Mfg. & Electrical Supply Co.**, 10228 Michigan Ave., Dearborn, Mich.

For more data circle MD-39, page 191

Overload Release Clutch

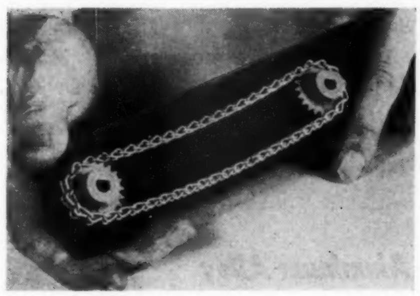
Universally applicable overload release clutch is fitted with an instantaneous trigger action and is designed to allow machines to resume operation at the exact cycle point of release. The clutch is dust

and dirtproof, and features permanent lubrication. Standard models cover a wide speed and shaft size range, however, special sizes can be ordered to specifications. Made by **Overload Release Clutch Co.**, Dept. E, 1162 Stuyvesant Ave., Irvington, N. J.

For more data circle MD-40, page 191

Miniature Chain

Available in sizes from 21 gage having 82 links per foot to 16 gage with 34 links per foot, this small chain is said to provide a quiet, economical, efficient and trouble-free method of drive or control. Regular or high tensile types are made to precision standards which

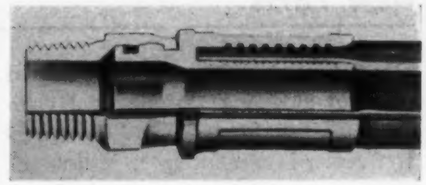


virtually eliminate all variation even in chain as fine as 82 links per foot. Made by **Hodell Chain Co.**, 3942 Cooper Ave., Cleveland 3, O.

For more data circle MD-41, page 191

Synthetic Hydraulic Hose

This flexible hose with full flow fittings for application in medium-high pressure hydraulic systems is said to be the first nonmetal-reinforced hose to offer working pressures in the 1800 to 2300 psi range. This is accomplished by employing a tough compar tube and high tensile synthetic fiber-braid carcass. Compar also renders the hose impervious to hydraulic oils. In addition, the all-synthetic construction makes the hose assemblies fatigue-proof. As further evidence of this, the assemblies were subjected to the tough Military Standard impulse test required for aircraft hose. Even though not re-

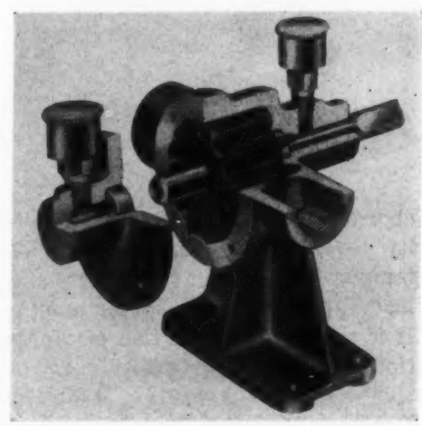


quired, the hose was maintained at 160 F during the tests. It is reported that there was no effect after 147,000 shock impulses. These lines are being offered in ¼, ⅜ and ½ in. ID. Physical pull strengths for these are 900, 2500 and 3500 pounds respectively. It is made by **Resistoflex Corp.**, Belleville 9, N. J.

For more data circle MD-42, page 191

General Purpose Pump

An impeller design that improves priming qualities and increases pump output with the same power is an outstanding feature of this pump. The impeller, molded in a tough durable rubber, has 12 vanes which result in greater displacement and greatly improve pump



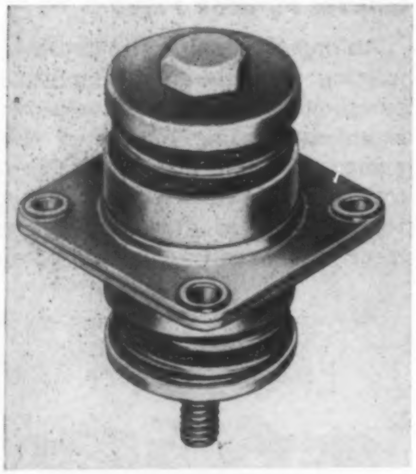
priming. Sharp lateral wiping edges on each vane minimize friction by reducing impeller drag, produce less unit wear and improve mechanical efficiency of the pump up to 15 per cent. Another advantage of the multivane impeller is the even continuous flow of liquid realized with the design. The pump is available in three models—½, ¾ and 1 in. port sizes. Pump case is rustproof bronze. A stainless steel shaft, replaceable unit-type seals and straight-through ports are used. Materials and design permit pumping of kerosene and fuel

oil. Dirty water can be pumped without clogging. Made by **Hypro Engineering Inc.**, 404 N. Washington Ave., Minneapolis 1, Minn.

For more data circle MD-43, page 191

Vibration Absorbing Mount

Developed to cope with extreme vibration and acceleration characteristics in guided missile applications, model 9302 double-acting all-metal Met-L-Flex mount is also adaptable to other uses. Features include nonlinear deflection

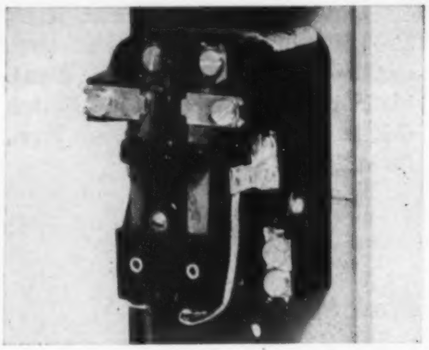


characteristics, high damping ability to withstand any condition of temperature and environment, and load ranges of 1 to 7 lb per mount. It is suitable for isolation of vibration and shock for positive, negative and radial loads. Made by **Robinson Aviation Inc.**, Teterboro, N. J.

For more data circle MD-44, page 191

General Purpose Relays

The CR2790-E series general purpose relays may be used as starters for small ac motors where motors have sufficient overload protection and are suitable for application in electronic equipment and air conditioners. Heavy silver contacts and braided shunts provide long life. Conversion units are available for metal and compound-base backmounting, base receptacle and re-



lay jack-plug applications, and explosion-proof enclosures for dust-tight and class 1, group D service. Relays are rated from 6 to 300 volts; 60, 50, and 25 cycles, plus dc; and 10-amp continuous contact rating. Contact arrangements include double-pole single-throw, double-pole double-throw, and single-throw double-break. Manufactured by **General Electric Co.**, Schenectady 5, N. Y.

For more data circle MD-45, page 191

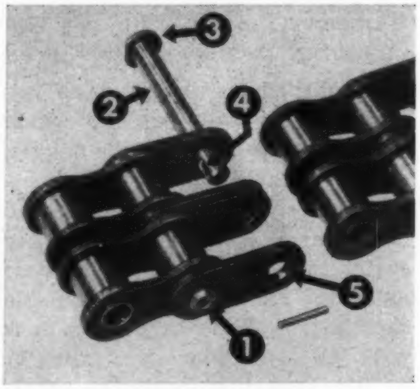
Aluminum Alloy

K155, a new magnesium-type aluminum sheet alloy for general metalworking reqiurements has the same general mechanical properties as the manganese-type alloy 3S but is expected to replace it in many applications because of advantages deriving from its magnesium content. As compared with 3S, K155 is well suited for taking anodized finishes as it shows less tendency toward structural streaking and has a clearer, lighter color after processing, that is free of the yellowish grey tinge of 3S or the brownish tinge of 52S. It closely matches the color of anodized 63S extrusions. With a nominal magnesium content of 0.9 per cent, K155 in the as-received condition does not have the dark coloration characteristic of the high magnesium-containing alloys such as 52S. K155 is available in plate, flat sheet, coil and circle form in all tempers in the H1 and H3 groups. The H1 temper series is used for applications requiring relatively simple forming; the stabilized H3 series is recommended where moderate or severe forming is involved. Mechanical property limits, availability and tolerances are the same for K155 as for 3S. Made by **Kaiser Aluminum and Chemical Sales Inc.**, 1924 Broadway, Oakland 12, Calif.

For more data circle MD-46, page 191

Riveted Roller Chain

Long wearing characteristics and ease of assembly and disassembly are combined in improved Baldwin assembly riveted roller chain. It is supplied in ASA standard and heavy series, in 1 through 2½-in. pitch sizes in multiple widths, and is offered in 10-ft lengths with



Baldwin-Rex single pin coupler. Spacing of couplers varies with chain size, but they are placed in the assembled chain so that any length can be made up without cutting rivets or damaging parts. Made by **Chain Belt Co.**, Baldwin-Duckworth Div., Milwaukee 1, Wis.

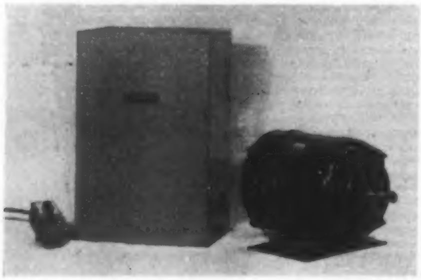
For more data circle MD-47, page 191

Variable Speed Control

Electronic variable speed motor control systems up to and including 3 horsepower models use special terminal block transformer construction, special cabinet design for wall or bench mounting, and plug-in assemblies for critical components. Electrical design utilizes the motor series field as a torque signal source in a patented feedback circuit to provide constant system

torque capability. Controlled acceleration on many models allows use in applications of high inertia loads without placing excessive torque load on materials in critical processes.

Fingertip controlled models for manual speed selection at remote

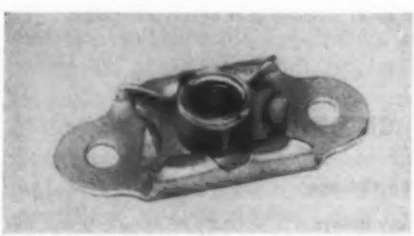


points are available, as well as a line of automatic Servospeed systems for use in various industrial operations or processes. Automatic tension control, constant surface speed windup, and other automatic models are available, as well as memory type systems operating from magnetic tape recordings to follow previously determined time-speed sequences. Besides the many standard models, specialized designs to particular customer specifications are available from the custom design section of Servospeed Div., **Electro-Devices Inc.**, 4-6 Godwin Ave., Paterson, N. J.

For more data circle MD-48, page 191

Floating Anchor Nuts

Produced from tempered spring steel in 10-32 and ¼-28 thread sizes, the F-5000 series of floating anchor nuts offers complete interchangeability with fixed type anchor nuts and complies in every respect with AN-N-5b and AN-N-10a specifications. Full 1/16-in. radial

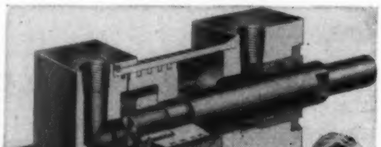


float facilitates assembly and eliminates alignment problems. In the 10-32 size, torque-out values in excess of 80 lb-in. and push-out values in excess of 400 lb are attained. In the ¼-28 size, torque-out values in excess of 120 lb-in. and push-out values in excess of 500-lb are attained. The nuts can be used at temperatures to 550 F. Normally furnished cadmium-plated, but available in other finishes for special applications from the **Kaynar Mfg. Co.**, 820 East 16th St., Los Angeles, Calif.

For more data circle MD-49, page 191

Hydraulic Cylinders

Packing cartridge unit of this hydraulic cylinder is easily and quickly changed by removing the snap ring, usually without dismounting the cylinder. Cartridge

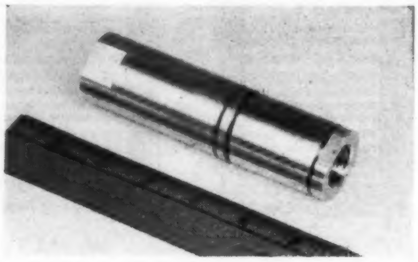


is machined from leaded bronze alloy and provides double bearing surfaces for the piston rod. Line of 2000-psi cylinders includes 11 models in 11 bore sizes, built to JIC standards. Photo shows cross-section of cylinder and inset shows the packing cartridge. Made by **S-P Mfg. Corp.**, 12415 Euclid Ave., Cleveland 6, O.

For more data circle MD-50, page 191

Relief Valve

Composed of less than a dozen parts, the Anco Relief Valve maintains pressure of oil, water or chemicals at a predetermined setting despite changes in demand. At the same time it protects pumps by allowing flow through the pump which, since it is always a direct flow back to the reservoir, acts as a coolant for the pump. Valve seat is formed of two hardened steel pieces which are ground and lapped for long life. Normally, the Anco valve is preset at the factory to pressure specifications. Made in an in-line envelope with either internal or external adjustments and locking, or in a side-port envelope with external adjustment and lock. Maximum free flow through valves is

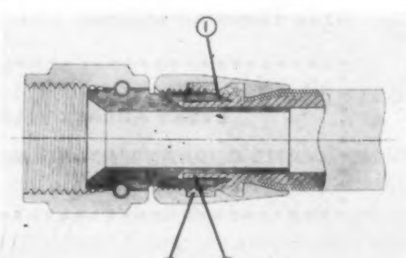


equivalent to that of the pipe used. Produced in both aircraft and standard pipe sizes, it is very light—for example, weight of the in-line type relief valve for ½ in. aircraft tubing is approximately 12 ounces. Made by **Anco Inc.**, One Baker St., Dept. 28-B, Providence 5, R. I.

For more data circle MD-51, page 191

Hose Fitting

Design of the Little Gem fitting for pneumatic and hydraulic hose reduces the compressive strain on the inner rubber or composition section of the hose tube, inhibiting flow of this material away from



the compression zone even in high temperature applications. Only the reinforcing braid of the hose is clamped, while the fluid-tight seal

Saved! **thousands of man-hours...thousands of bearings... thousands of dollars!**

ALEMITE OIL-MIST

the most efficient machine lubrication ever devised!

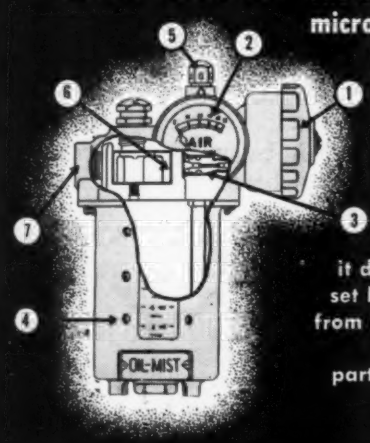
1 MULTIPLIES Bearing Life!

2 SLASHES Product Spoilage!

3 BOOSTS Machine Output!

It's almost unbelievable, but the record proves it's true! Oil-Mist has literally saved thousands and thousands of dollars. Even more important is the equally impressive record of increased production. Increases impossible without this completely new conception of lubrication. This increased production, increased bearing life and decreased product spoilage has paid for installation after installation in a matter of weeks.

This is a completely new, amazingly simple, system of lubrication. Easy to incorporate into new projects, just as easy to use in modifying existing designs. Oil-Mist applies a constant, clean, cool film of oil uniformly to working parts—groups of bearings, slides, chains, gears—wherever needed. And in the form needed—liquid, spray or mist. The lubricator has no moving parts . . . operates on compressed air, is completely automatic, completely foolproof!



HOW IT WORKS: The Oil-Mist Lubricator atomizes oil into microscopic particles which are carried in the air stream and distributed through tubing to bearings.

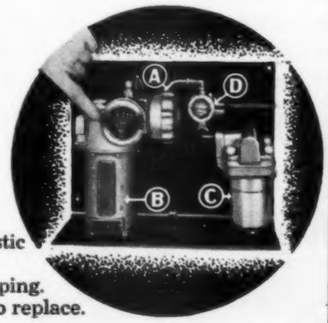
Oil-Mist airborne lubrication is accomplished this way: Compressed air entering the unit passes through air regulator (1) and air gauge (2). As this air passes through venturi (3) it draws oil from reservoir (4). Oil flow is set by knob (5). The mixture of air and oil from the venturi is thrust against baffle (6). Only the most minute, lighter-than-air particles are blown through outlet (7) into delivery line to lubricate bearings.

Alemite Oil-Mist offers these lubrication advantages

Automatic Lubrication • Continuous Lubrication • Eliminates Guesswork • Greater Safety Cuts Oil Consumption up to 90% • Extension of Bearing Life • Stops Oil Drippage Reduction of Bearing Temperatures • Manpower Savings • Elimination of "Down-Time" Reduction in Number of Lubricants • Protection from Contamination

LUBRICATOR SPECIFICATIONS:

- Oil-Mist outlet ½" fem. p.t. Air gauge registers to 50 psi. Operating air pressure – 5 to 20 psi.
- Air regulator **(A)** reduces from pressures up to 200 psi. Normal air consumption –.7 to 1.2 cfm.
- Range of oils handled – to 1,000 sec. (S.U.V.) @ 100°F.
- Oil reservoir **(B)** capacity 12 oz. (approximately 1 week supply). Intake filter screen – 70 mesh. Fill plug – ⁹⁄₁₆".
- Material – die cast aluminum body with nylon plastic window.
- Baffle-type water separator **(C)** —automatic self-dumping. Requires no manual attention – no filter elements to replace. Air inlet ¼" fem. p.t.
- Solenoid Control **(D)** starts system automatically when machine starts – foolproof.



delivers oil to bearings 3 ways!

1. Oil-Mist as Such. Most commonly applied to any type of anti-friction bearing – ball, roller or needle.

2. Oil in Spray Form. For open and enclosed gears and chains. Nozzle partially condenses mist so that it can be directed on to a concentrated area.

3. Oil-Mist Condensed. For plain bearings, slides, ways, vees, cams and rollers. In these applications, condensing fittings convert oil-mist to liquid oil.

A PRODUCT OF SW STEWART WARNER

Write Today! **This Oil-Mist Catalog and Engineering Data Book is FREE for the asking. Write now for your copy. Alemite, Dept. R-73, 1850 Diversey Parkway, Chicago 14, Illinois.**

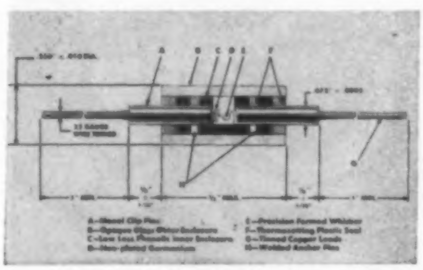
Alemite OIL-MIST Lubrication

is accomplished by separating the inner tube from the braid and using it as a lip seal in an annular chamber. Illustration shows (1) knife-like spur which automatically separates hose inner tube and reinforcing wire braid (2) reinforcing wire braid of the hose clamped between the nipple and the socket, and (3) hose inner tube seating in annular chamber to form a positive lip seal. Made by **Aeroquip Corp.**, Jackson, Mich.

For more data circle MD-52, page 191

Germanium Diodes

Germanium diodes, consisting of an inner housing of low-loss phenolic material over which an outer housing of glass is assembled, are resistant to moisture, shock and



vibration. Design affords electrical isolation from adjacent circuits. Retention of original whisker adjustment is maintained by welded anchor pins. All types are available from **International Rectifier Corp.**, 1521 East Grand Ave., El Segundo, Calif.

For more data circle MD-53, page 191

Oil Coolers

Use of longitudinal inner fins makes possible a more compact unit than has been possible previously. Oil flows through the patented Heat-X inner fins, which provide generous surface for cooling with minimum pressure drop. Various models are suitable for oil-side pressures up to 250 psi and temperatures up to 300 F. Water side is suitable for pressures up to 150 psi. The inner-finned element

is all copper and is removable for cleaning and replacement. Another feature is the use of synthetic rubber O-rings to provide positive seals and eliminate all packed and soldered joints. The entire unit can be disassembled for cleaning or replacement. Applications include: diesel and gasoline engines, hydraulic presses, turbines, gears, compressors, hydraulic couplings, torque converters, transformers, etc. Made by **The Heat-X-Changer Co.**, Brewster, N. Y.

For more data circle MD-54, page 191

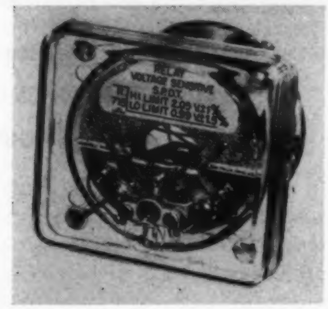
Corrosion Resistant Paint

NeoCoat is a new type of Neoprene anticorrosion coating. A catalyst and a retarder is included in the mix to eliminate necessity of mixing on the job. After application the retarder leaves with the solvent and the catalyst cures the coating into a synthetic thick film having chemical and abrasion-resistant properties. Effective in covering welds, beads, seams and edges where coatings most frequently fail. NeoPrime primer is applied before the NeoCoat is brushed or sprayed on surface. A true plastic, this coating is manufactured by **Pennsylvania Salt Mfg. Co.**, 1000 Widener Bldg., Philadelphia 7, Pa.

For more data circle MD-55, page 191

Meter-Relays

Moving-coil relays with self-locking contacts have factory adjusted accuracy within 2 per cent of the specified current or voltage for most values. Adjustment can be made at installation to improve the accuracy to better than 1 per cent. Operating on as little as 0.2-microamp, meter-relays are also made for 50 amp or for any intermediate value. In voltage ranges they offer a choice from 0.05-mv to 500 v. Higher values of voltage

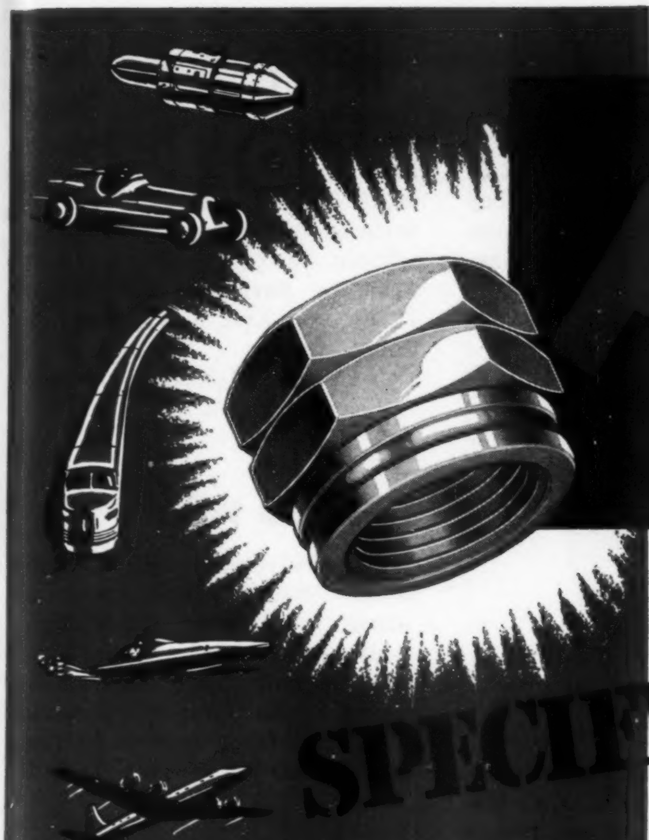


or current can be handled with external resistors. The sensitivity is adjustable at installation over a wide range. These relays can be used on alternating current with the addition of a rectifier. Crystal diodes or instrument type copper oxide rectifiers are suitable. Rectifiers can be supplied internally or they can be mounted outside the relay. There is approximately the same range of voltage and current in ac relays. Normal contacts have ratings up to 100 dc milliamperes for *make* only. Heavy duty contacts are rated 1 amp at 115 v ac or 32 v dc for *make* and *break*. Make by **Assembly Products Inc.**, P. O. Box 191, Chagrin Falls, O.

For more data circle MD-56, page 191

Self-Aligning Barrel Nuts

Designated Series SL 6, these light, compact self-aligning barrel nuts are designed for application where highly concentrated loads are encountered. The nuts consist of three pieces: a 75S-T6 aluminum barrel; a threaded steel bushing; and a steel retainer. The ears on the base of the threaded steel bushing float in the slotted section of the retainer, thus providing the self-aligning feature. The knurled retainer is pressed in place and punch staked at four places. Shape of the lightweight barrel distributes the load carried by the steel



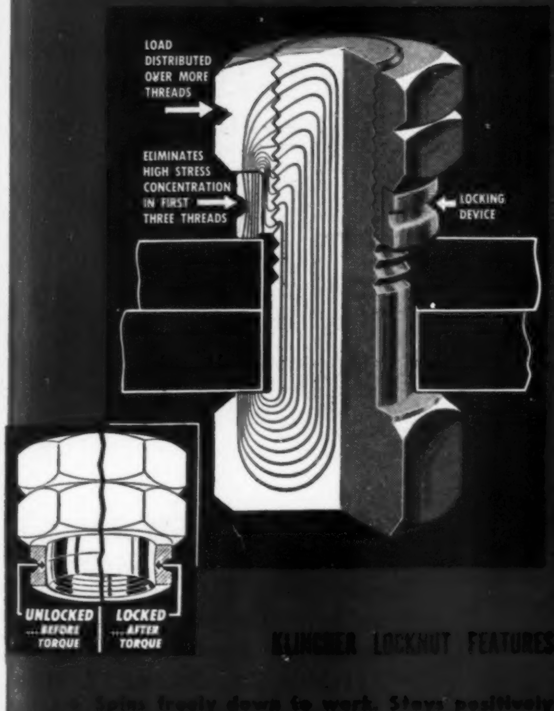
Klincher LOCKNUT

SPECIFIED

Wherever VIBRATION
HIGH TEMPERATURE up to 1600°
CORROSION
TENSILE STRENGTH
RE-USABILITY
are factors!

To meet the needs of designers for a single fastener combining all these stringent characteristics, Klincher Locknut was developed. Punishment meted out to bolt connections on today's jet and conventional aircraft engines, racing cars, Diesel engines and equally severe applications, demands a locknut like Klincher. Its one-piece handling offers many production-line and maintenance advantages.

Inquiries invited regarding specialized locknut problems. Write for data and experimental samples, giving size and application.

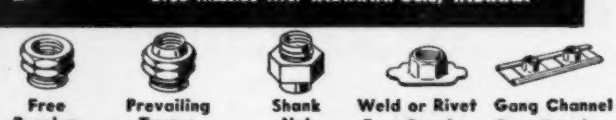


KLINCHER LOCKNUT FEATURES

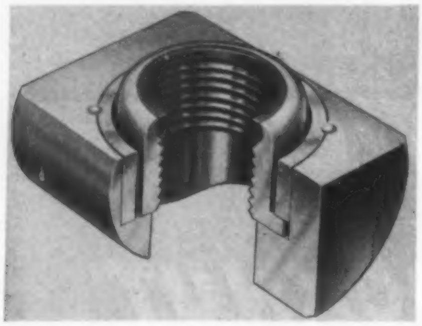
- Spins freely down to work. Stays positively locked.
- After breakaway, nut spins off freely, ready for reuse, without impairment of locking efficiency.
- Only one piece to stock and handle . . . saves time installing and removing.
- Ideal for standard and power wrenches.
- Manufactured in stainless steel and other materials.

Klincher LOCKNUT CORP.
2153 HILLSIDE AVE. INDIANAPOLIS, INDIANA

MEMBER INDUSTRIAL FASTENERS INSTITUTE



National Fine and National Coarse Threads.

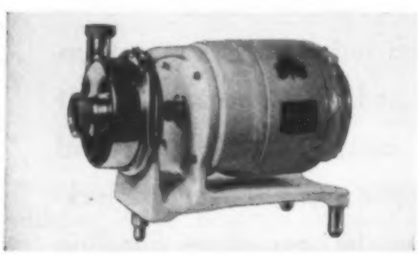


threads of the bushing into surrounding structure. Made with internal thread sizes from ¼-28 to ⅞-14 by **Shurlock Corp.**, 9010 Bellanca Ave., Los Angeles 45, Calif.

For more data circle MD-57, page 191

Centrifugal Pumps

Line of corrosion-resistant sanitary and industrial centrifugal pumps are available in a complete range of sizes and capacities up to 1250 gpm and up to 250 ft head at

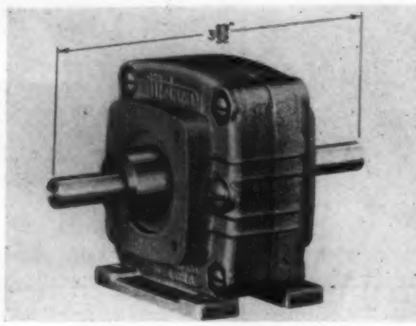


zero gpm. Pumps have screw impellers, carbon seals, lightweight heads, and Tri-Clamp casing construction. Designed to operate at 1800 and 3600 rpm, these units can pump food products, beverages, drugs, chemicals in solution, brine, and paint. Made by **Tri-Clover Machine Co.**, Kenosha, Wis.

For more data circle MD-58, page 191

Bantam Size Speed Reducers

Small, yet powerful, speed reducers are now available in the "Bantam" Series 10 line. Nominally rated at 0.1-hp, these speed reducers contain high quality, hardened steel, 48 pitch gears. Over 600 different ratios are available. Only 2 7/16 in. high and 2⅜ in. wide, these speed reducers can be used in very close quarters; they may be used for chart drives, computers, oscillographs, servomechan-

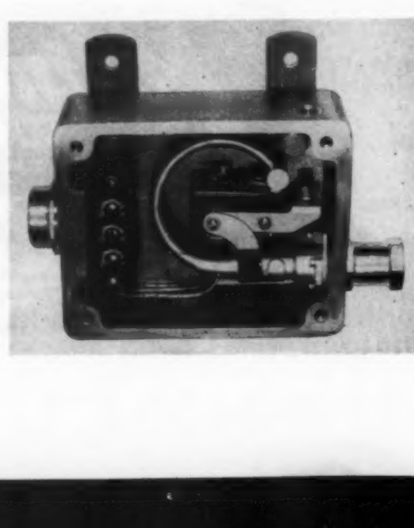


isms, etc. The 5/16-in. diameter in-line shafts make them easy to use in existing equipment. Anti-friction bearings throughout plus a splash type oil bath insure long, dependable, trouble-free performance. Made by **Metron Instrument Co. Inc.**, 432 Lincoln St., Denver 9, Colo.

For more data circle MD-59, page 191

Pressure Switches

Six classes of proof pressures, 500, 1500, 3000, 4500, 6000 and 12,000 psi, are provided in model 314 Meletron pressure switches. They provide accurate sensing of system pressures over an adjustable range of 15 to 10,000 psi and are suitable for liquid, gas or water systems. They are available with external adjustment if setting requires frequent changes. Variable actuation value ranges from 10 to 2000 psi, depending on switch class.



Actuation can be on increasing or decreasing pressure automatically reset by snap action of switch. Model 304 skeleton switches with same operating characteristics, but without case, are also available. Made by **Barksdale Valves**, 1566 E. Slauson Ave., Los Angeles 11, Calif.

For more data circle MD-60, page 191

Solenoid Air Valves

The 3500 series of ¼-in. in-line solenoid-operated spring-return air valves is available for straightaway and three-way application in both normally open and normally closed types. Conduit connection is separate from cover to facilitate wiring. All types have a 3/16-

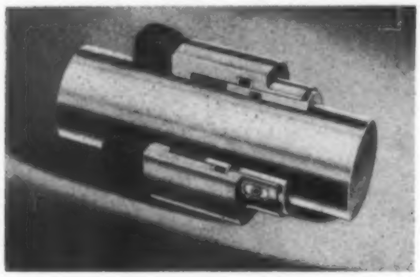
in. orifice area and are available in any voltage and cycle requirement. Recommended working pressure is from 20 to 150 psi. Made by **Mechanical Air Controls Inc.**, 15311 W. 11 Mile Rd., Detroit 37, Mich.

For more data circle MD-61, page 191

O-Ring Seals

Design simplicity is a feature of these O-ring mechanical seals for rotating shafts on centrifugal pumps handling chemicals, petroleum products, edible liquids, pulp liquors, and other liquids. Type O

unbalanced seal is made for pressures up to 200 psi, while type OB balanced seal, illustrated, handles pressures of 1000 psi and over. Both can be supplied with single



or multiple springs, in either pin-drive or sleeve-drive construction. Type O seals are made for packing spaces 5/16-in. and larger, type OB for 3/8-in. spaces and larger. O-rings can be Teflon, Buna-N, Neoprene or silicone. Made by **Garlock Packing Co.,** Palmyra, N. Y.

For more data circle MD-62, page 191

Gear Pumps

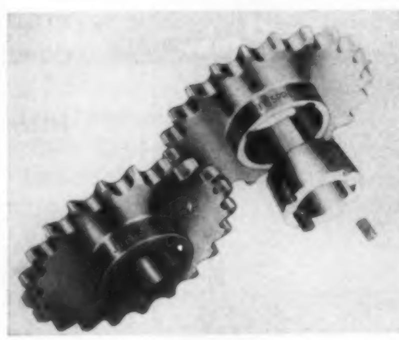
Designed for such service as pressure lubricating, oil circulating, transfer and filtering systems, and lift and replenishing systems, the LB series of gear-type hydraulic pumps will handle most liquids having a substantial oil base. Three capacities of 2.5, 3.5, and 4.5 gpm at 1800 rpm and 100 psi are available, and pressures or speeds in excess of these may be used satisfactorily in selected applications. The series includes two types of pumps, type LBP for face mounting or wet sump application, and

type LBS which includes a shaft seal and is recommended for direct, gear or belt drives. Type LBS is also available in a 115/230-v motor-pump combination. An internal relief valve, adjustable for pressures between 50 and 200 psi, is optional. Made by **Webster Electric Co.,** 1900 Clark St., Racine, Wis.

For more data circle MD-63, page 191

Taper-Lock Sprockets

Taper-lock principle eliminates need for boring sprockets to fit shafts and regrinding or turning of shafts to get tight fit. Bore sizes

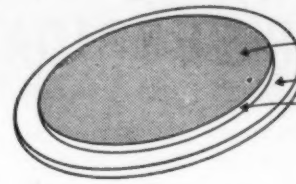


range from 1/2 to 3 in. in 1/16-in. increments. Pitches from 40 to 100 are available. Bushings may be used with sprockets having different numbers of teeth and same sprocket may be used on various shafts by changing bushings. Made by **Union Chain and Mfg. Co.,** Sandusky, O.

For more data circle MD-64, page 191

Filtration Material

Completely resistant to all strong acids, caustics, oxidizing agents and common organic solvents, Kel-F plastic filtration medium can be used at up to 300 F, has 900 psi tensile strength and elasticity modulus of 1800 psi. Mean pore opening is 15 microns. Flow capacities range downward from 500 cfm of air per square foot of filter surface, at 8 psi differential pressure, and from 110 gpm of water per square foot at 10 psi

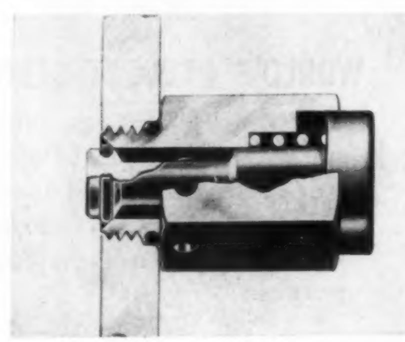


differential pressure. Material is available in disks up to 12 in. diameter, in thicknesses of 1/16 and 1/8-in. and with chipproof contiguous solid Kel-F edges. Made by **Porous Plastic Filter Co.,** 30 Sea Cliff Ave., Glen Cove, N. Y.

For more data circle MD-65, page 191

Pressure Release Valve

Designed to allow rapid escape of air pressure in systems, bleed air from fluid systems and similar uses, a new manually operated release valve has only four working parts. Design assures dead-tight sealing and has 9/16-18 threads, permitting normal installation with



either an O-ring, as shown, or with gasket seal. Valve is held tightly closed by spring when manual pressure is released. It is pressed to open and released to close. Made by **James-Pond-Clark,** 2181 E. Foothill Blvd., Pasadena 8, Calif.

For more data circle MD-66, page 191

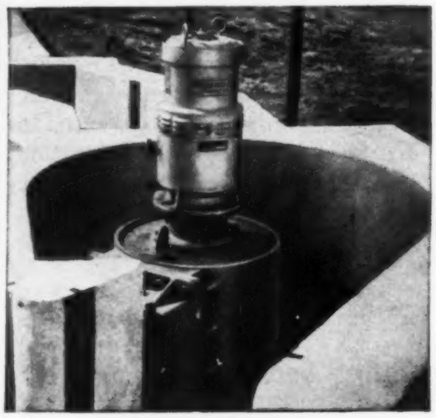
Snagproof Nut

By shielding the points of screws, the type J Speed Nut protects upholstery and fabrics against snagging, as well as offering protection against personal injury or product damage by scratching. Special nut

Advertisement

Weather Proof Motors for Continuous Outdoor Duty On Chicago Sewage Equipment

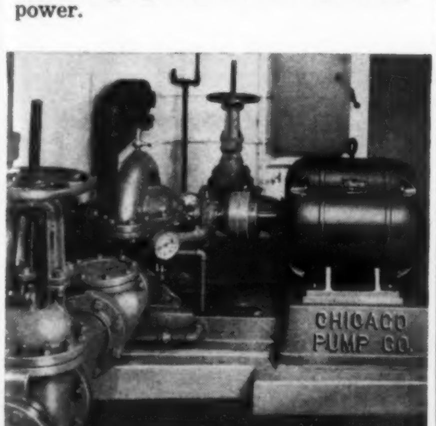
Chicago Pump Company, manufacturers of Sewage Treatment Equipment, use Star-Kimble weather proof gearmotors exclusively on COMMINUTORS. These units are usually installed outdoors without weather protection and must provide continuous operation in all weather conditions. More than 2500 COMMINUTORS with Star-Kimble gearmotors are installed in Sewage Treatment Plants throughout the world. The unfailing operation of the COMMINUTOR is essential to the successful operation of the Sewage Treatment Plant.



Chicago Pump Company's COMMINUTOR, powered with Star-Kimble motor with special full weather proof construction.

Chicago Fire Pump

Another Chicago Pump Company application of Star-Kimble motors is for Fire Booster Pump service. These pumps provide water pressure for sprinkler systems and standpipes. Chicago Fire Pumps are driven by Star-Kimble open drip proof protected motors, ranging from 40 to 100 horsepower.



Chicago Fire Pump, driven by Star-Kimble 40 H.P. squirrel cage motor.

STOP FAST FAST STOP FAST FAST STOP
START SMOOTHLY SMOOTHLY START

FREQUENTLY FREQUENTLY REVERSE FREQUENTLY FREQUENTLY
POSITIVELY POSITIVELY HOLD POSITIVELY POSITIVELY

with Star-Kimble Brakemotors

Motor *and* connected load come to a split-second stop—because of the extra-large brakelining area of a Star-Kimble Brakemotor.

Motor starts smoothly—without drag—because small air gap assures instantaneous brake release.

No plugging needed for rapid, repeated reversals. Frequency of reversals can often be *tripled,* compared with conventional plugging methods.

And loads are held positively by the brake that holds its grip through millions of cycles—with little or no adjustment.

Every Star-Kimble Brakemotor is an integral, space-saving unit with a sturdy shaft that's common to motor and brake. Bearings and brakelinings last longer. One manufacturer—one responsibility.

For information on design and service features, write for Bulletin B-501-A

STAR-KIMBLE

MOTOR DIVISION OF MIEHLE PRINTING PRESS AND MFG. CO.

201 Bloomfield Avenue Bloomfield, New Jersey

has a rigid integral flap curving up and back over the point of the screw where it comes through the nut. Made by **Illinois Tool Works,** Shakeproof Div., St. Charles Rd., Elgin, Ill.

For more data circle MD-67, page 191

Pyrometer Controllers

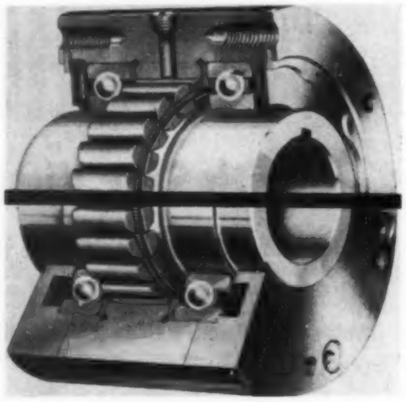
Close temperature control of furnaces, ovens, plastic molds and other heating equipment is provided by line of Free-Vane electronic pyrometer controllers. Units serve to actuate relays, electric contactors, solenoid valves and motor

valves. Control system is based on frequency modulation principle, a millivoltmeter mechanism and unit plug-in construction. Basic model can be used for low-open, high-open and low-high control. Model with extra plug-in unit proportions current input for straight-line control of many heating appliances. A double control unit is offered to provide low-open-high or low-normal-high control. All models can have thermocouple protection. Available in wide selection of temperature ranges from 0-400 to 0-3000 F. Controllers are made by **Bristol Co.,** Waterbury 20, Conn.

For more data circle MD-68, page 191

Over-Running Clutches

A centrifugal throw-out sprag assembly now makes it possible for Formsprag ball-bearing over-running clutches to operate at speeds exceeding 3000 rpm. Assembly eliminates possible rubbing between sprags and the inner race. Construction of these clutches consists of outer housing, inner race, full complement of sprags and energizing springs. The sprag principle provides an infinite number of gripping positions and eliminates backlash. Maximum torque capac-

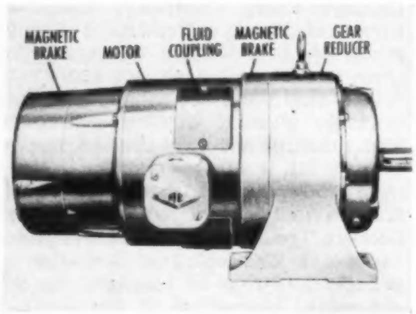


ity and long life because of changing contact points are advantages. Made by **Formsprag Co.,** 23601 Hoover Rd., Van Dyke, Mich.

For more data circle MD-69, page 191

Motor & Magnetic Brake

Smooth acceleration and deceleration is provided by the Motoreducer in addition to completely stationary holding action when stopped. Unit consists of a magnetic brake, motor with internal fluid coupling, second magnetic brake, and gear reducer. Typical of the uses include crane travel drives or any heavy load that would tend to produce a pendulum motion with quick stops. In operation the energized motor transmits power to the load through the fluid coupling. Load

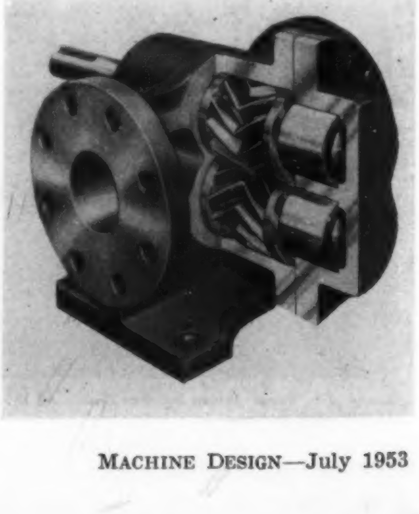


is decelerated through the fluid coupling by the magnetic brake mounted on armature. Second brake mounted between coupling and gear reducer can be used to decelerate or hold the load stationary. Manufacturer is **Reuland Electric Co.,** Alhambra, Calif.

For more data circle MD-70, page 191

Rotary Gear Pump

Designed for positive displacement of lubricating liquids, this Hydrex rotary gear pump handles fluids and semifluids with viscosities from 32 to 250,000 SSU. Capacities are 1 to 500 gpm at 300 psi for continuous duty and 500 psi for intermittent service. Body and head are the only castings used, and the two precision rotor gears are only moving parts. Shafts are hardened and ground at the heavy

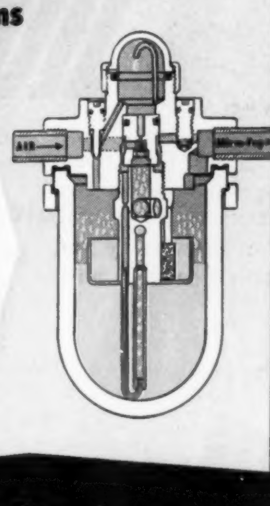


NOW

Norgren MICRO-FOG

for
multiple-bearing applications
to

- **cut costs**
- **allow greater flexibility in design**



lubricate up to 40 bearings with one Micro-Fog Lubricator

Norgren oil-fog lubricators are proved by the fact that they are standard on most types of equipment where the best of lubrication is required for high speed spindles, air cylinders and valves.

Now, the advantages of Micro-Fog Lubricators have been extended to applications on bearings, gear boxes and small air-operated devices. It produces an extremely fine and uniform air-borne Micro-Fog at low air flow; transmits it over greater distances than previously possible; distributes it evenly thru as many as 40 outlets; provides exact control and uniform rate of oil feed.

Write for new catalog sheet No. 485.

PIONEER AND LEADER IN OIL FOG LUBRICATION FOR 25 YEARS

• Lubricators • Valves • Regulators • Hose Assemblies • Filters

C. A. Norgren® CO.

3442 So. Elati St., Englewood, Colorado

duty roller bearings; only one stuffing box is used. All connections over 2 in. are flanged. Machined surfaces are ground for easy piping and aligning. Made by **Sier-Bath Gear & Pump Co.,** 9252 Hudson Blvd., North Bergen, N. J.

For more data circle MD-71, page 191

Miter Gear Boxes

Compact and powerful, these miter gear units are made in two sizes of $\frac{1}{3}$ and 1 hp at 1800 rpm. Each size has both a two-way and three-way shaft extension with

choice of five mounting facings. The $\frac{1}{3}$-hp unit has a static torque of 250 lb-in. while that of the 1-hp unit is 750 lb-in. Made by **Crown Gear, Inc.,** 290 West St., Keene, N. H.

For more data circle MD-72, page 191

Foot Switch

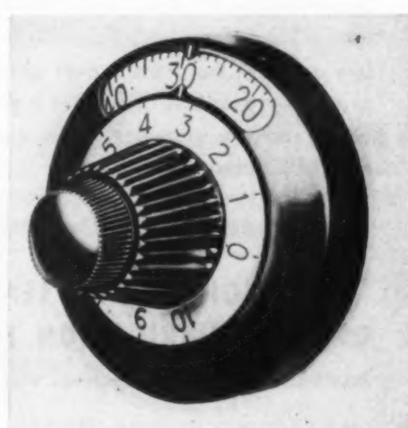
Type MA foot switch uses a long-life SPDT limit switch. Access to internal switch terminals is greatly facilitated by simply removing the front end of the new two-piece top casting. Actuating treadle built into the top of the MA foot switch requires unusually light foot pressure and permits fast operation with minimum fatigue. Cable enters the switch through the front by means of a BX connector or a rubber grommet, as specified. Mounting is by means of holes in the housing. The red treadle and

machine gray design are harmonious, and in keeping with industrial design for office and plant use. Made by **General Control Co.,** 1200 Soldiers Field Rd., Boston 34, Mass.

For more data circle MD-73, page 191

Multiple Turn Dial

A high-precision ten-turn counting dial, capable of being friction-locked in any fixed position by turning an externally-mounted knob, may be used on any multiturn device having ten turns or less. Composed of two concentrically mounted dials, one for increments of 0.01-turn and the other for counting turns; thus contact position can be indicated to an indexed accuracy of one part in one thousand. The

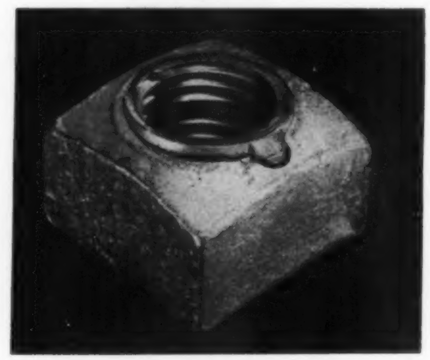


turning knob is attached rigidly to the shaft on which the incremental dial and shaft coupling are mounted, eliminating any chance for backlash. It can be friction-locked securely against accidental turning by means of a locking knob mounted on the external end of the turning knob; dials are also locked. Made by the **George W. Borg Corp.,** Janesville, Wis.

For more data circle MD-74, page 191

Galvanized Locknut

This locknut, which can be hot-dip galvanized, has a square body and holds its position on the bolt by means of a locking insert. The locking insert is a slightly elliptical, heat-treated retainer ring of spring

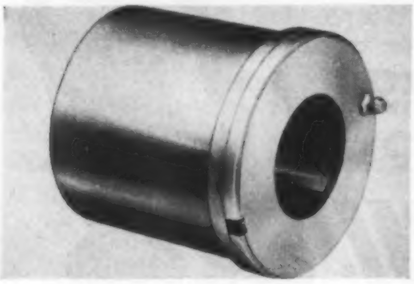


steel. When the nut is applied, the bolt forces the retainer into the round, causing the retainer to grip the bolt tightly. Bolt tension is not required, and the locknut can be removed and replaced any number of times without destroying its locking power or damaging the bolt. It will hold its position wherever it is "wrenched." This galvanized nut may be used where alkali, salt water or other corrosive elements are present. It is made in sizes from $\frac{3}{8}$ to $1\frac{3}{8}$ in. by the **Security Locknut Corp.,** North Ave. & 15th Ave., Melrose Park, Ill.

For more data circle MD-75, page 191

Needle Bearing Bushing

Providing an absolute seal against entrance of dirt, coolants, and grit, needle bearing pilot bushings are available with inside diameters of 0.257 to 2 in. Maximum bar support is provided because long overall lengths from $1\frac{1}{2}$ to $3\frac{1}{2}$ in. are available. Quills can be

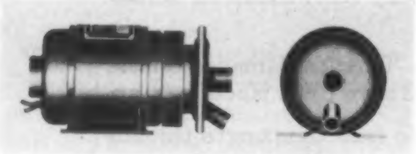


adapted to receive keys or keyways. Bearings can be disassembled without removing housing. Protection against thrust as well as radial loads is provided. Manufactured by **J. G. Jergens Co.**, 11106 Avon Ave., Cleveland 5, O.

For more data circle MD-76, page 191

Midget Centrifugal Pump

The model MU midget centrifugal pump, for handling liquids where capacity requirements range from 0 to 6 gpm of free flow to 6 ft of head, has been added to the Pioneer pump line. Designed for continuous or intermittent operation, the pump is directly connected



to a 60 cycle, 115 v, single phase motor in the 1/50 to 1/20-hp range. Made of brass, stainless steel or Monel metal, and is 5 in. in diameter and 9 in. long. Inlet pipe is ⅞-in. OD and outlet pipe is ⅝-in. OD. Available from **Detroit Harvester Co.**, Pioneer Pump Div., 19657 John R St., Detroit 3, Mich.

For more data circle MD-77, page 191

Miniature Connectors

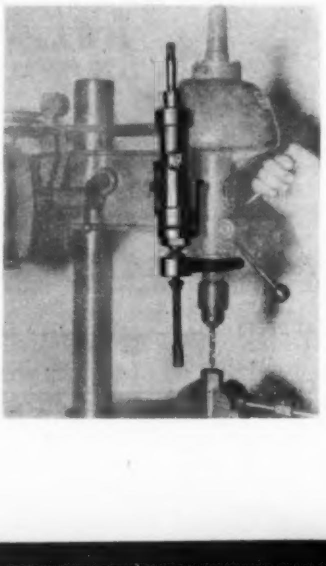
For use on miniature sealed and unsealed instrument switches, relays, transformers, amplifiers and other equipment, the U group of connectors is presently made in three and six-contact arrangements; single and 12-contact types will be available. Pin and socket insert types are available, as are hermetically sealed units for solder applications and locknut units for mechanical or solder applications. Contacts are for 5-amp current, with a minimum flashover of 1700 v dc. Shells are steel, finished in chromate-dipped cadmium plate. The three-contact plug weighs 0.0088-lb and is 1 1/16-in. long

with a maximum diameter of 17/32-in. Made by **Cannon Electric Co.**, 3203 Humboldt St., Los Angeles 31, Calif.

For more data circle MD-78, page 191

Feed Controller

The DC-50100 Power Check feed controller will control feed rates of both manually and power operated machines in addition to other types of straight-line motion. External adjustments permit control during all or part of the stroke in addition

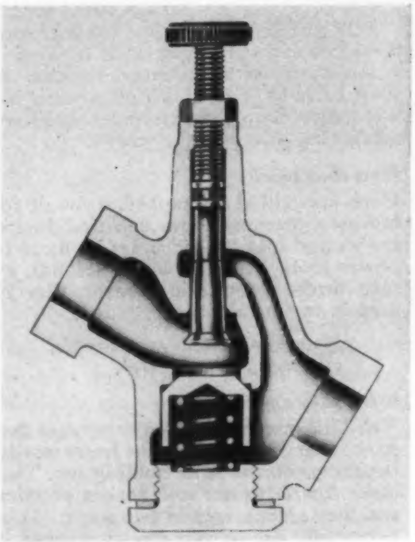


to variation of the feed rate. Control is achieved by the resistance of a fluid being forced through a small orifice. Available with strokes of 2, 4, 6 or 8 in. from **National Pneumatic Co. Inc.**, 125 Amory St., Boston 19, Mass.

For more data circle MD-79, page 191

Speed Control Valve

Split-second timing of piston movement by positive control of air flow is provided by this adjustable speed-control valve. It can be mounted in any position between operating valve and one or both ends of a cylinder to provide air flow adjustment. Flow-controlling orifice can be adjusted from practically zero to wide open. Valve is



offered with ¼, ⅜, ½, ¾, 1 and 1¼ in. IPT. Made by **Ross Operating Valve Co.**, 120 E. Golden Gate Ave., Detroit 3, Mich.

For more data circle MD-80, page 191

Fractional Horsepower Motors

Two special-service Form G fractional-horsepower motors, for applications requiring moderately high starting torque, are rated at ½ and ¾-hp at 1725 rpm. The motors can be mounted in any posi-

To get better prints... faster

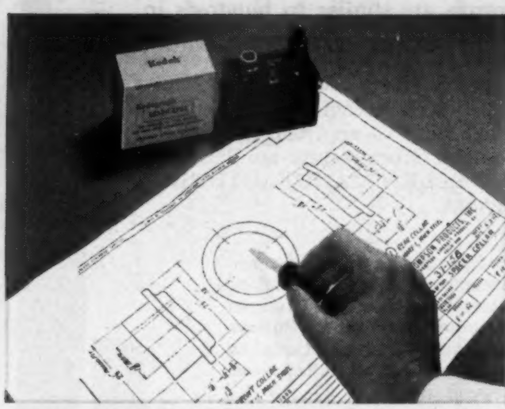
Tapco Plant, Thompson Products, Inc. uses Kodagraph Autopositive intermediates in print production.

The Tapco Plant knows full well that illegible shop prints pave the way for costly reading errors; knows, too, that Kodagraph Autopositive intermediates are low-cost insurance against such a possibility.

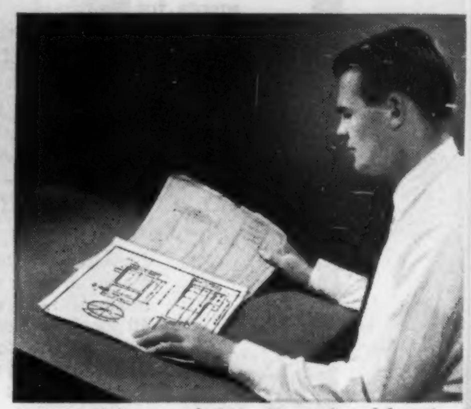
It therefore reproduces its more critical and complex jet and valve drawings on Kodagraph Autopositive Paper Translucent and thus obtains—quickly and easily—sparkling "masters" for print-making which have dense photographic black lines on an evenly translucent, premium-quality paper base; *which* will produce highly legible whiteprints time after time at *stepped-up* machine speeds.

Extremely fine-detailed drawings are reproduced on Kodagraph Autopositive Film, which captures the faintest detail . . . keeps close lines from "filling in". . . and produces top-quality photographic intermediates which have extremely fast print-back speeds.

No negative step . . . no darkroom handling. Kodagraph Autopositive Paper and Film are handled in exactly the same manner . . . produce positive photographic intermediates *directly*. First, they are exposed in one of the Tapco Plant's direct-process machines—or in a photocopy unit. Then, they receive standard photographic processing. A fast, convenient room-light operation all the way. And no new equipment needed.



Autopositive intermediates save creative drafting time. Tapco Plant makes necessary changes in its basic designs without costly redrafting by (1) making Autopositive prints of the original drawing; (2) removing the unwanted detail from the Autopositive reproduction with eradicator fluid. Then, the draftsman has only to add the new detail . . . and a print-making master is ready. *One* which will produce highly legible prints without confusing "ghost" images in the eradicated area.



Autopositive reclaims "unprintables." Many old drawings that have lost line density or are soiled or torn are transformed into print-making masters by reproducing them on Autopositive Paper or Film. Stains and crease marks are dropped out . . . weak detail is made more legible—saving hours of redrafting. Autopositive Paper is also used to duplicate a variety of office records, non-translucent vendor prints, etc.

Kodagraph Autopositive Materials

"THE BIG NEW PLUS" in engineering drawing reproduction

Write today for a free copy of "Modern Drawing and Document Reproduction." It gives complete details on the revolutionary line of Kodagraph Reproduction Materials, which you, or your local blueprinter, can process conveniently, economically.

MAIL COUPON FOR FREE BOOKLET

EASTMAN KODAK COMPANY

Industrial Photographic Division, Rochester 4, N.Y.

19

Gentlemen: Please send me a copy of your illustrated booklet giving the facts on Kodagraph Reproduction Materials.

Name______________________ Position______________________

Company______________________

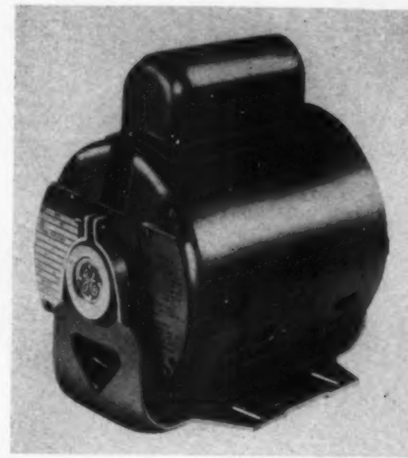
Street______________________

City______________ Zone________ State______________

Kodak

TRADE-MARK

New Parts

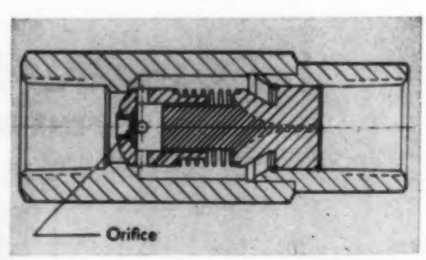


tion and are easily reconnectable at the terminal board for 115 or 230 v ac. They are smaller and lighter in weight than the corresponding general-purpose ratings, according to the manufacturer. Starting currents conform to NEMA standards. Made by **General Electric Co.,** Fractional-Horsepower Motor Dept., Schenectady 5, N. Y.

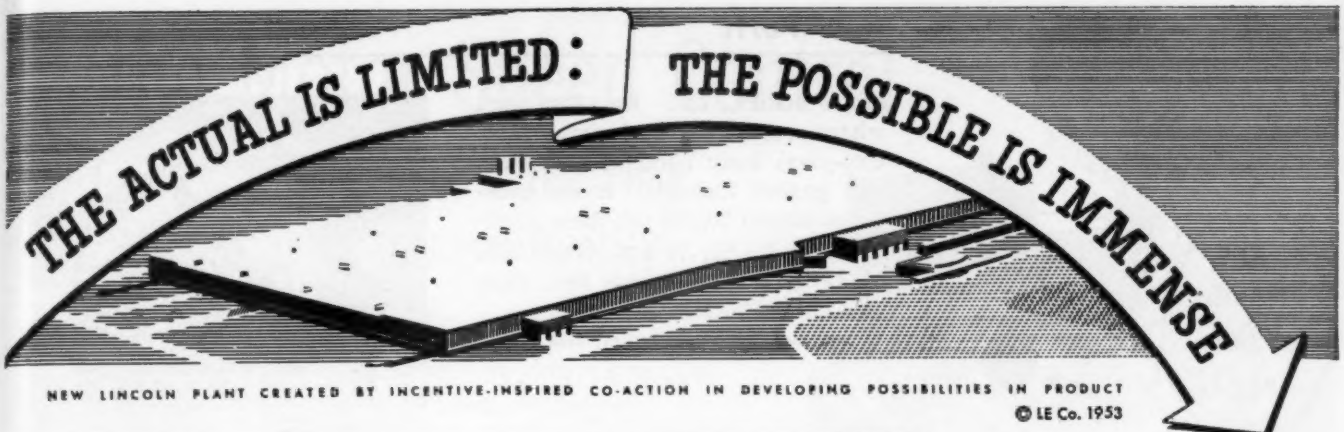
For more data circle MD-81, page 191

Restrictor Valves

These nonadjustable restrictor valves, designed to regulate the speed of a cylinder in one direction, are check valves with an orifice in the check poppet. Free flow is permitted in one direction by the check valve; reverse flow closes the check and must pass through the orifice. The size of the orifice is made to



suit the application. Low pressure drop in the free-flow direction is obtained by using standard check valve parts. The valves have large seats and ample flow areas. Normally furnished with a one-piece copper-brazed steel body; alumi-



NEW LINCOLN PLANT CREATED BY INCENTIVE-INSPIRED CO-ACTION IN DEVELOPING POSSIBILITIES IN PRODUCT

© LE Co. 1953

WELDED DESIGN REDUCES TRUCK ASSEMBLY WEIGHT 38%

A 38% weight reduction resulted when Detroit Automotive Products Corp., Detroit, redesigned their Twin-Booster Axle Assembly for all-welded fabrication. Weight was reduced, a vital factor in truck manufacture, from the 74 lbs. of the original design to 46 lbs.

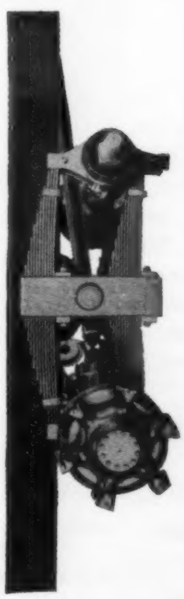


Fig. 3. New welded *spring hanger assembly strengthens Twin-Booster Axle Assembly, yet weight is cut 28 lbs.*

New welded design cut cost of the assembly *66%*. Costly machining is virtually eliminated.

Original assembly consisted of an expensive, machined casting, two sheared and bent plates, and two small castings. The first redesigned "spring hanger assembly" eliminated the large casting. Lincoln welding advisors suggested further weight reductions and simplifications shown in final design, Fig. 2.

Latest ideas for simplifying conversions from cast iron to welded steel construction are presented in Lincoln Bulletin, "Elements of Weldesign". Available to designers and production engineers by writing on your letterhead to Dept. 1106.

NEW DESIGN IN WELDED STEEL CUTS COST REDUCES WEIGHT

Fig. 1. Original design *with castings weighed 74 lbs. Required extensive machining. Assembly had 5 costly components.*

Fig. 2. Present design *of welded "spring hanger assembly" weighs only 46 lbs. Less machining required. Eliminates need for expensive patterns.*

HERE'S HOW

THE LINCOLN ELECTRIC COMPANY

CLEVELAND 17, OHIO

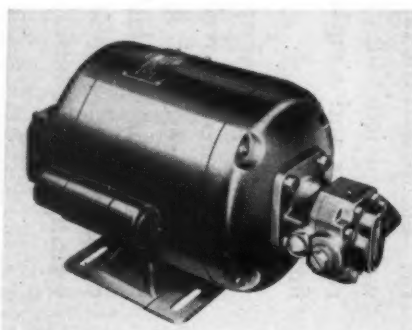
THE WORLD'S LARGEST MANUFACTURER OF ARC WELDING EQUIPMENT

num bodies can be furnished where weight saving is important. One-piece body eliminates gaskets and gasket leakage. Burst pressures exceed 10,000 psi. Sizes available are ¼, ⅜, ½ and ¾-in. IPT. Made by **Fluid Controls Inc.**, 1284 N. Center St., Mentor, O.

For more data circle MD-82, page 191

Motor-Pump Unit

This unit combines a motor with a rotary gear pump. Motors are furnished in three frame sizes, covering a range from ⅛-hp to 1½ hp, including all popular intermediate hp ratings. Motors are available in either the open or to-



tally enclosed type, for 25, 50, or 60 cycle operation at 110, 220, 440 or 550 v. Equipped with a low pressure pump, this combination provides 500 psi maximum pressure. With a high pressure pump, a constant pressure of 1500 psi or an intermittent pressure of 3000 psi is available. Made by the **John S. Barnes Corp.**, 301 S. Water St., Rockford, Ill.

For more data circle MD-83, page 191

Air Regulator

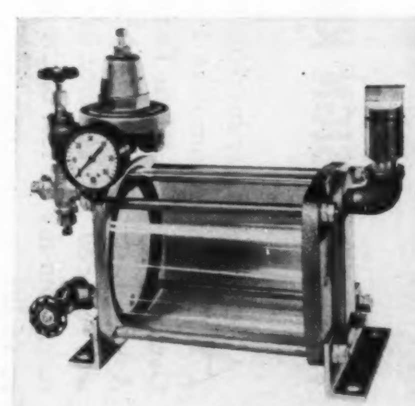
Capable of handling a large volume of compressed air with a minimum of pressure drop, the type HAA regulator has a capacity of 80 cfm at 100 psi line pressure. Two models, each basically the same with the exception of inlet and outlet connection arrangements are manufactured. Sensitive

control and low pressure losses are achieved by using air pressure against a regulating diaphragm. Maximum pressure which can be regulated is 135 psi and maximum line pressure is 300 psi. The new unit is about 3 in. wide and 4½ in. high. The control knob is grooved for a sure grip and is made of plastic. All exposed metal surfaces are plated. Made by the **DeVilbiss Co.**, 300 Philips Ave., Toledo 1, O.

For more data circle MD-84, page 191

Air Operated Oil Dispenser

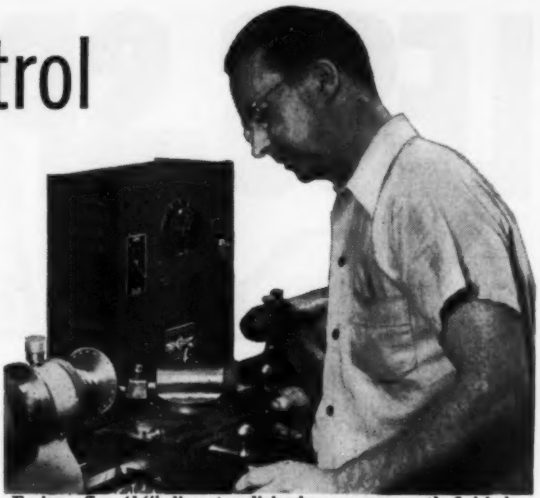
Where gravity feeding is not suitable, oil and other fluids can be fed at any desired rate by air pressure to elevated, distant and



inaccessible points by this DHP dispenser. Practical for many applications including spray oiling, unit is designed to serve as central reservoir of single-line low-pressure

the Motor Speed Control

that Cuts Lathe Time and Improves Quality



Facing off a 4½" diameter dial where very smooth finish is required. The Type 1702-A Variac Motor Speed Control enables the operator to slow lathe speed as larger diameter is worked. Maintaining cutting-feet-per-second constant avoids chatter.

Variac MOTOR SPEED CONTROLS are COMPLETE units for operating DC Motors from AC Lines, and feature:

- ★ Low Speed
- ★ Selenium Rectifiers for Ruggedness
- ★ Good Regulation
- ★ No Pulsating At Low Speeds
- ★ Easy Installation and Maintenance
- ★ Wide Speed Range
- ★ High Starting Torque
- ★ Rapid Starting and Stopping
- ★ Dynamic Braking

If you need a speed control which will increase the everyday versatility of your lathe and enable you to meet the most stringent production requirements . . . the Variac Motor Speed Control is the answer. This smoothly operating control makes lathe work much more efficient. In production, it enables quality work to be turned out rapidly and reliably. Variac controls are available in three sizes for 1/15th to ¾ hp motors.

LATHE OPERATORS who use these Controls REPORT THESE TIME-SAVING FEATURES:

★ *Variac Speed Controls eliminate need for changing step pulleys or gears — twist of a knob gives any speed from 5 or 10 rpm to rated*

★ *Instant Starting — no waiting for gears to mesh or engage as in mechanical controls — rapid reversing where needed*

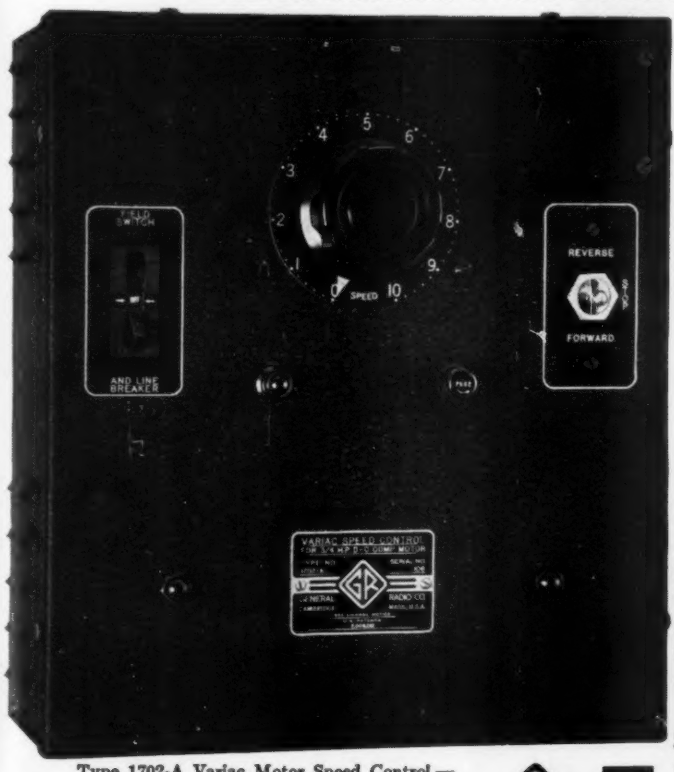
★ *Instantaneous Braking Saves Production Time — permits sudden stops, eliminating tool overruns when under power feed or when cutting screw threads*

★ *Infinite Selection of Speeds — machine is slowed or speeded for optimum operation by flick of the wrist — in turning or facing operations, speed can be varied continuously as diameter of stock changes*

★ *Power at Few RPM — considerably more power at slow speeds makes this control invaluable in reaming and many other applications*

★ *Less Wear and Tear On Tools — speed can be set to suit material being worked or tool being used*

★ *Tapping in Slow Motion — with Variac Speed Control, lathe can be run at few rpm while tap is slowly run into drilled piece — eliminates necessity for measuring depth of drill hole on tap and carefully watching distance to which tap is advanced — motor stalling when tap is all the way in prevents tap breaking or spoiling of threads*



Type 1702-A Variac Motor Speed Control — plugs into regular 115v a-c power lines — provides continuous control of ⅓ and ¾ hp d-c motors — unit is completely self-contained — price $230, less motor

Variac Motor Speed Controls are amazingly efficient devices for the control of everyday machine operations including precision drilling — grinding — toroidal wire winding and screw cutting. They are time tested, highly recommended tools widely used in machine shops and large-scale factory production departments throughout the country.

GENERAL RADIO Company®

275 Massachusetts Avenue, Cambridge 39, Massachusetts, U. S. A.

90 West St. NEW YORK 6 · 920 S. Michigan Ave. CHICAGO 5 · 1000 N. Seward St. LOS ANGELES 38

SEND IN COUPON FOR MORE INFORMATION

General Radio Company, 275 Massachusetts Ave., Cambridge 39, Mass.

Please send me a copy of the VARIAC Motor Speed Control Bulletin:

Name______________________________
(578)

Company Name________________________

Street______________________________

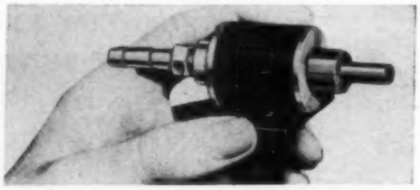
City______________Zone______State________

air-operated oiling system discharging regulated amount of oil by means of adjustable sight feed needle valves. Operation is on 80 psi line. Sizes with up to 2-gal Lucite reservoir for floor or wall mounting are furnished. A 0 to 60 psi air pressure gage and regulator is standard. Made by **Oil-Rite Corp.**, 2376 Waldo Blvd., Manitowoc, Wis.

For more data circle MD-85, page 191

Small Air Cylinder

With 1-in. stroke and equal ram pressure at any given point in the stroke, Micro model air cylinder can be used where size and space limitations are factors. It delivers

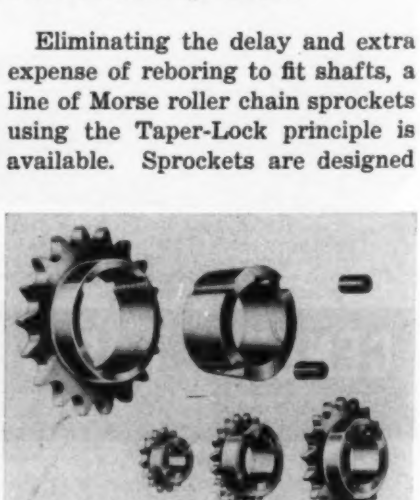


75-lb thrust on a 100-psi line. Any number of the solid brass cylinders can be operated by a single control valve. Made by **Air-Mite**, 2653-T W. Lake St., Chicago 12, Ill.

For more data circle MD-86, page 191

Roller Chain Sprockets

Eliminating the delay and extra expense of reboring to fit shafts, a line of Morse roller chain sprockets using the Taper-Lock principle is available. Sprockets are designed



Now Available

AUTO-LITE

Silicone-insulated Wire

Auto-Lite Silicone-insulated Wire is immediately available for use on civilian products — where extreme temperature ranges occur. Check these advantages of Auto-Lite Silicone-insulated Wire for both high- and low-voltage applications.

- **Withstands extreme temperature ranges — from −60°F to 400°F.**
- **Good retention of electrical properties.**
- **Highly resistant to bacteria and fungi.**
- **Extremely good weather aging qualities — does not become brittle.**

Auto-Lite has the facilities available to accommodate your biggest demands for Silicone-insulated Wire. Inquire today!

THE ELECTRIC AUTO-LITE CO.
Wire and Cable Division
Port Huron, Mich.   Hazleton, Pa.

* * *

Tune in "SUSPENSE!" ... CBS Television Tuesdays

AUTO-LITE

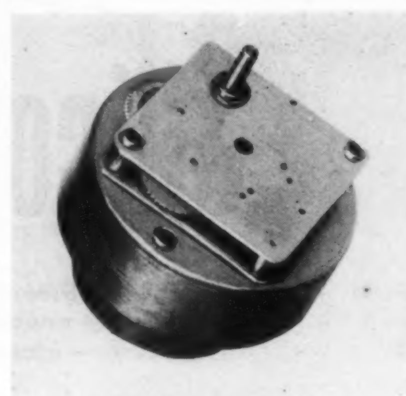
Wire and Cable

for easy, quick installation and removal, yet grip shafts with the firmness of a shrink fit. Having no flanges or protruding parts, the sprockets require no more space than standard sprockets. Fit any roller chain manufactured to American Standards No. 40 through 100. Come in pitchs of ½, ⅝, ¾, and 1¼ in., and number of teeth range as high as 112. Available from **Morse Chain Co.,** 7601 Central Ave., Detroit 10, Mich.

For more data circle MD-87, page 191

Small Motors

Available with speeds from 10 revolutions per hour to one revolution per month, series S-200 synchronous motors feature an additional exposed low speed gear train and will handle a continuous load up to 1 lb-in. and an intermittent load of 5 lb-in. Units have 2-in. diameter and 1⅝-in. depth. Made by Bristol Motor Div., **Vocaline Co. of America Inc.,** 205 Coulter St., Old Saybrook, Conn.



For more data circle MD-88, page 191

Stainless Steel

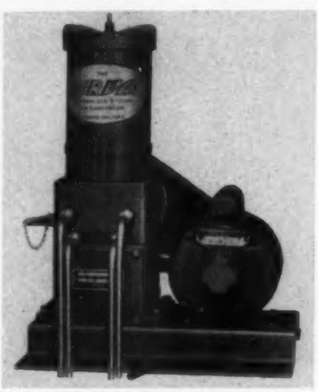
An austenitic alternate for 18-8 stainless offers good cold working properties, good weldability, and good ductility. Produced with manganese, chromium and less than one per cent nickel as principal alloying elements, this steel is now

being produced in substantial quantities as strip, and most other forms are also available. Developed because of the restricted usage of nickel, in an effort to obtain an austenitic stainless steel with less than one per cent nickel, this steel can be used where straight chromium steels are unsatisfactory as substitutes for 18-8. Made by **Allegheny Ludlum Steel Corp.,** 2020 Oliver Bldg., Pittsburgh 22, Pa.

For more data circle MD-89, page 191

Lubricant Pumping Unit

Designed primarily for automatic lubrication of single machines, this DC-20 unit accurately meters grease or oil to bearings. An integral time clock controls the



lubricating cycle, assuring lubrication at the proper intervals. A ¼-hp ac motor drives the pump through a V-belt. Lubricant reservoirs are available in three sizes to hold from 4½ to 12½ lb of grease or 2½ to 6½ qt of oil. Made by **Farval Corp.,** 3249 E. 80th St., Cleveland 4, O.

For more data circle MD-90, page 191

Worm Drive Assembly

Designed for general industrial machinery, this small right-angle worm drive is for application to shafting from ⅜ to 1 in. diameter. Compact unit has no external projection of the worm beyond the small housing. Drive incorporates

Part of the Engineering and Research Laboratory

Aerial view of Glenville Plant

FELT problems are welcomed here —AND SOLVED!

American makes felt in actually hundreds of different types, each having carefully-controlled characteristics. Felt, you see, is not just felt, but is an engineering material, which can be, and should be, selected and specified as closely as any other material.

American is keenly progressive and has a vast knowledge of all types of natural and synthetic fibres. Felt is now engineered into the various end uses. Our knowledge is freely available to you, through our sales staff, or from the Engineering and Research Laboratory.

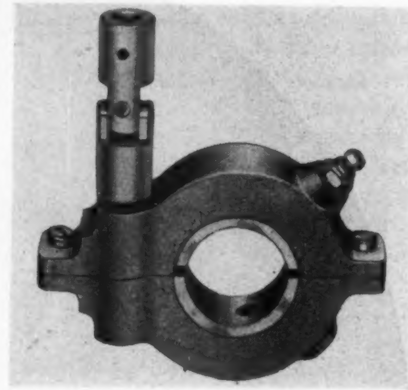
How important it is to obtain the right felt is illustrated by the case of a customer who insisted on "saving money" by buying a felt which we insisted was not suitable for the application. In the end, the saving produced a loss, and the customer, having learned the hard way, now relies upon our advice. You can avoid such trouble by bringing your needs for felt to American. Tell us what the felt is to be used for, whether in a process or a product, and we will help you select the right type.

And remember, American operates cutting shops in Glenville, Conn., Detroit, Mich., San Francisco, Calif., which can quickly produce cut felt parts, ready for assembly, to speed your output.

American Felt Company

TRADE MARK

GENERAL OFFICES: 22 GLENVILLE ROAD, GLENVILLE, CONN. SALES OFFICES: New York, Boston, Chicago, Detroit, Cleveland, Rochester, Philadelphia, St. Louis, Atlanta, Dallas, San Francisco, Los Angeles, Portland, Seattle, San Diego, Montreal.—PLANTS: Glenville, Conn.; Franklin, Mass.; Newburgh, N. Y.; Detroit, Mich.; Westerly, R. I.—ENGINEERING AND RESEARCH LABORATORIES: Glenville, Conn.

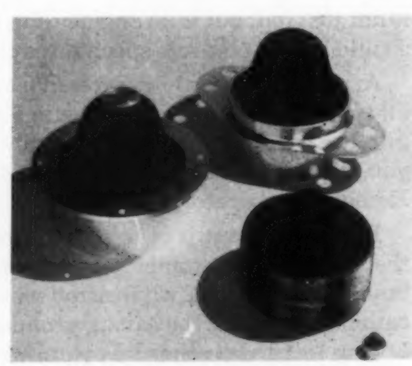


a locking ball drive so that the entire assembly can be placed on the shafting, closed in by bearings without disassembly or drilling of the shaft. Alemite grease fitting is provided for lubrication. Reduction ratio is one turn of universal joint to ten turns of the driving shaft. Made by **Durant Mfg. Co.,** 1933 North Buffum St., Milwaukee 1, Wis.

For more data circle MD-91, page 191

Vibration Isolator

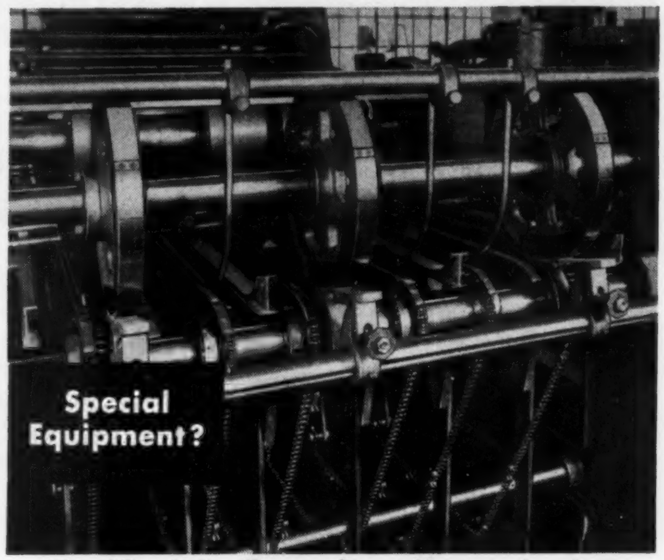
Shock and vibration protection during both shipment and use is afforded to delicate equipment, electronic instruments, aircraft engines and similar units by Cohrlastic DS nonlinear mounts. Small vibrations are absorbed with little or no deflection of movement of the equipment. At high loadings, the



mount serves as a cushion to take the impact energy and return the equipment unharmed to its original position. It has little tendency to develop resonant frequencies and is available in a wide range of

What Kind of Machinery Do You Make?

Diamond Roller Chain Drive (multiple strand) on high speed Rock Drill Rod Machine.



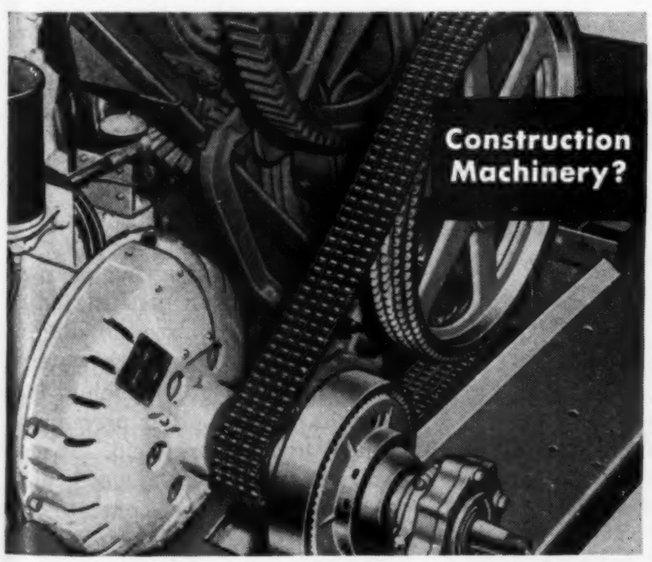
Adaptability, high uniform quality, positive operating characteristics make Diamond Roller Chains first choice for countless applications.

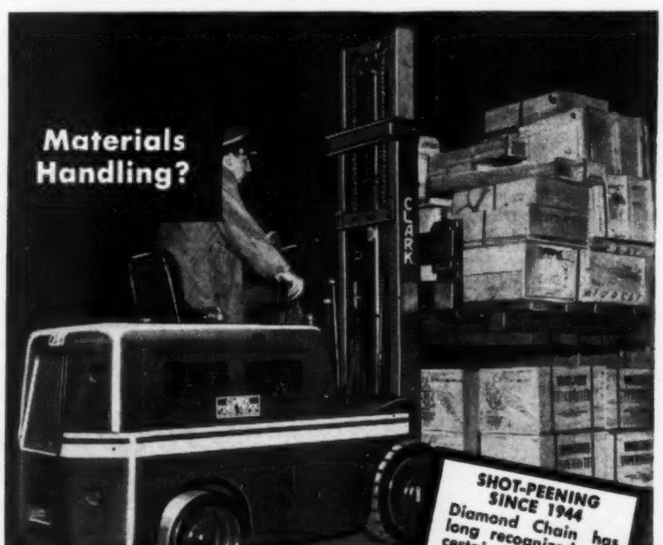
DIAMOND ROLLER CHAINS

Insure Top Operating Performance

For heavy machinery of all kinds, Diamond Roller Chains prove their great reserve strength and ability to withstand shock loads and tension.

For long hours of continuous service, the reliability and proven uniformity of quality of Diamond Chains make them outstanding choice in the materials handling field.





SHOT-PEENING SINCE 1944

Diamond Chain has long recognized that certain types of internal stressing of chain parts would increase fatigue resistance. To this end, link plates have been specially processed and chain rollers and other parts have been shot-peened since 1944.

In our files there are invaluable data, field records and other information to help solve your power transfer problems. Our engineering staff is at your service.

DIAMOND CHAIN COMPANY, Inc.

Where High Quality is Tradition

Dept. 435, 402 Kentucky Ave., Indianapolis 7, Ind.

Offices and Distributors in All Principal Cities

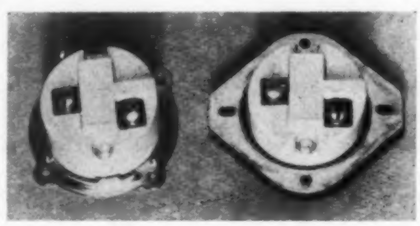
DIAMOND ROLLER CHAINS

sizes from ½ to 1000-lb rated loads. Made by **Connecticut Hard Rubber Co.**, 407 East St., New Haven 9, Conn.

For more data circle MD-92, page 191

Thermal Switches

Available in surface and recessed models, these are fixed temperature, thermally actuated, snap-action switches. Contacts are actuated by a snap-action bimetal blade of entirely new design which results in extremely fast response to temperature changes. A large

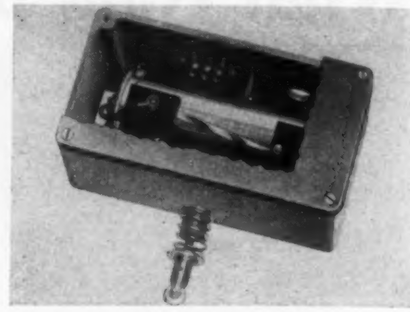


contact gap permits high current ratings in small compact units. Ceramic housing provides full 250 v spacing of terminals without use of separate mounting insulators. Units are supplied normally open or normally closed as required. Temperature ranges to specification. Manufactured by **The Wilcolator Co.**, 1001 Newark Ave., Elizabeth, N. J.

For more data circle MD-93, page 191

Plunger Potentiometers

Constant cutting speed machine tool drives, winder drives, processing machinery and dancer-roll systems are potential applications for bulletin 68 plunger potentiometers, intended for introduction into industrial electronic control devices. The vitreous-enameled resistance element and precious-metal sliding contact are protected by an oiltight enclosure with external mounting holes. Operating plunger, with its roller type cam follower, requires only ½-in. linear movement for complete traverse of the 10,000-



ohm potentiometer. Make by **Ward Leonard Electric Co.**, Mount Vernon, N. Y.

For more data circle MD-94, page 191

Silicone Rubber Parts

Molded and extruded silicone rubber shapes are now available. These shapes have excellent resistance to temperature extremes and an inert surface which minimizes sticking. They are not subject to change due to contact with weather, oils, common chemicals and weak acids and are odorless and tasteless. Available from Republic Rubber Div., **Lee Rubber & Tire Corp.**, Youngstown 1, O.

For more data circle MD-95, page 191

Thermoplastic Cement

F-88 is a quick-setting thermoplastic cement for general industrial use. It requires no air for setting (it will set under water); sets hard, ready for finishing, in a very short time; and is permanent, even in hydrochloric acid. It may be used to join metals, wood, plaster, porcelain, stone, concrete and glass.

F-88 is prepared by the user by mixing powder and liquid to make a thick creamy mix. By the choice of available liquids, it is possible to bond materials permanently in as little as five minutes. Available in a complete range of colors and in standard container kits up to 100 pounds from **American Consolidated Dental Products Co.**, Industrial Div., P. O. Box 2015, Philadelphia 3, Pa.

For more data circle MD-96, page 191

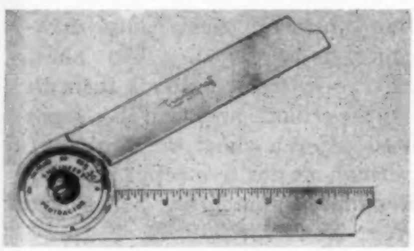
ENGINEERING

DEPARTMENT

EQUIPMENT

Adjustable Protractor

Any number of angles ranging from 0 to 180 deg can be drawn using this new protractor without turning, sliding, or moving unit out of position at any time. Upper arm swings up to any desired angle and can be locked by an adjustment knob. Magnifier built-in over registration mark provides clear hairline adjustment of settings. Sharply defined 1 and 5-degree black division lines are printed on white



background through 180 deg arc. Pertinent mathematical formulas are printed on reverse side of calibrated dial.

Clear plastic unit has 5-inch scale imprinted on lower arm. The instrument is 2 in. wide and 7 in. long when set at zero deg; set at 180 deg overall length is 12 in. Available from **Way-Mac Mfg. Co.**, 8112 Melrose Ave., Los Angeles 46, Calif.

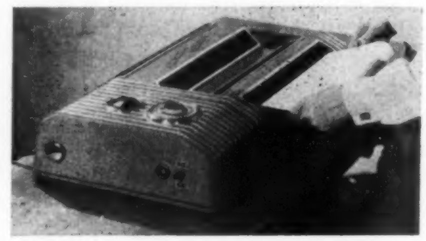
For more data circle MD-97, page 191

Photocopy Machine

Finished photo-exact, positive copies of any record regardless of type or color can be made by the Transcopy Duplex in less than a minute without developing, washing, fixing or drying. Photocopies of originals up to 14½ in. wide in any length can be handled. Two-

in-one unit contains both a printer and a processor. Machine operates after being plugged into any 115 v ac electrical outlet with no darkroom needed.

A negative sheet is placed face to face with an original to be copied and inserted into the front exposure slot of the unit. When the two sheets automatically emerge the original can be released immediately. The exposed negative paper is then placed face to

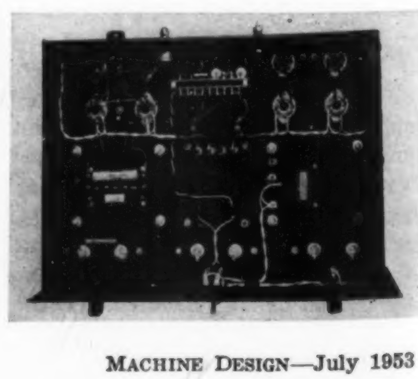


face with a sheet of Transcopy Transfer paper and inserted in the front slots of the developing section until an electrically driven roller is engaged. The sheets emerge in a few seconds from the delivery slot, and can be separated after a short wait, giving a permanent positive copy. Made by **Remington Rand Inc.**, 315 Fourth Ave., New York 10, N. Y.

For more data circle MD-98, page 191

DC Power Supply

Providing 300 v dc power from a 115 v ac, 50 or 60 cycle source, the Eastgap Model Two adds inherent reliability to these specifications. Ripple combined with noise and jitter is $\frac{1}{3}$ millivolt or less. Impedance level is 1/20 ohms or less down to zero frequency

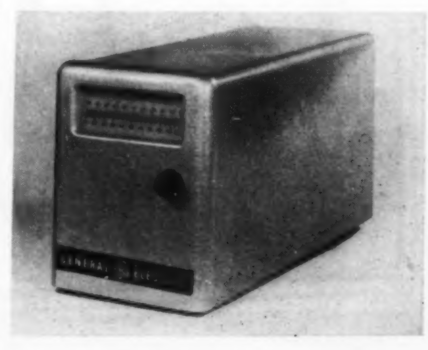


throughout load range. A 15 microfarad oil condenser is incorporated directly across the output terminals. Conservatively load rated at 0 to 1000 milliamperes. Built with hermetically sealed transformers and chokes having grain-oriented cores. Extra heavy filtering by using large oil capacitors affords maximum front-end efficiency before regulator. Amplifier has high gain as well as high stability and balanced input. Designed for bench or rack mounting. Made by **Eastgap Company,** 285 Columbus Ave., Boston 16, Mass.

For more data circle MD-99, page 191

Portable Galvanometer

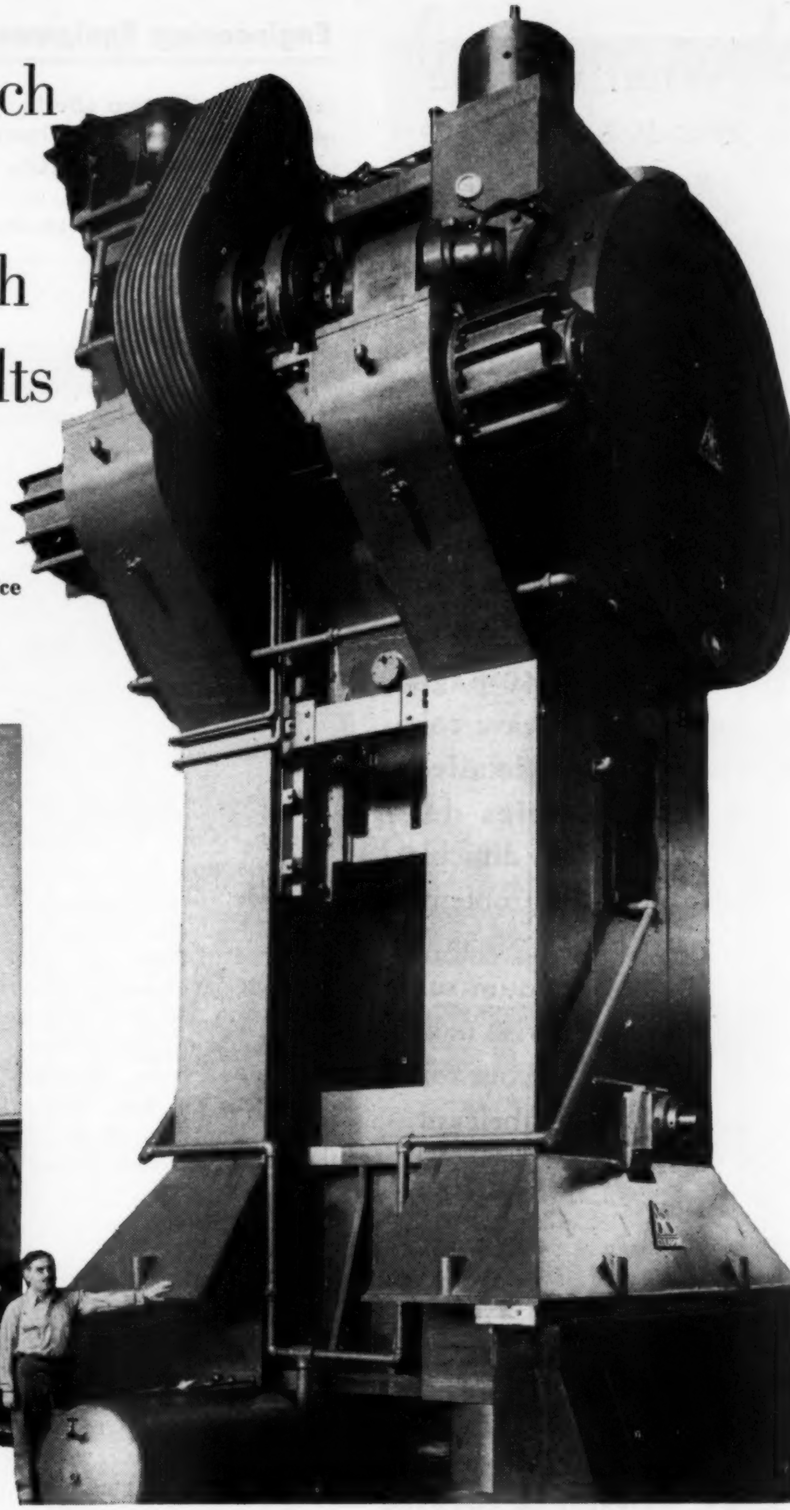
A new line of high-sensitivity portable galvanometers for measuring low values of current or voltage has been announced. Higher sensitivity is made possible by the use of Alnico magnets. Tilted scale with hairline index and spotlight allows readings to be taken



quickly and accurately. The scale is marked 50-0-50 and 0-100 in 1-mm divisions and is made of a translucent compound.

The instrument is designed so that the entire coil and suspension element can be removed from the magnetic assembly and replaced by another system of different sensitivity, without the necessity of remagnetizing or disturbing external connections. The galvanometer element and optical system are mounted in a lightweight metal case approximately 6½ by 8 by 16 inches, which is shock-mounted to

Controlling the punch of mighty heavyweights...with U.S. Rainbow® V-Belts



A Complete Drive Service
Multiple V-Belts
F. H. P. Belts, Sheaves
Flat Belting and Belting
Special Purpose Belts

Note the U.S. Rainbow V-Belts in multiple drive on these presses. The monster at the right has a capacity of up to 3000 tons.

These powerful presses, used for forming steel, exert tremendous forces. The V-Belts that transmit this power must be capable of handling great shock loads. The belts designed by United States Rubber Company for these presses have the right strength and dependability for the job. Each belt has the unique Equa-Tensil Cord Section that scientifically distributes the load to insure efficient pull with minimum stretch. Protective jackets increase the grip while keeping out dirt and prolonging wear. For large or small drives, you can depend on the quality of U.S. Rainbow V-Belts. Order from your jobber or contact any of the 25 "U.S." District Sales Offices, or write to address below.

U.S.RUBBER
SERVING THROUGH SCIENCE

"U.S." Research perfects it...
"U.S." Production builds it...
U.S. Industry depends on it.

UNITED STATES RUBBER COMPANY
MECHANICAL GOODS DIVISION · ROCKEFELLER CENTER, NEW YORK 20, N. Y.

Hose • Belting • Expansion Joints • Rubber-to-metal Products • Oil Field Specialties • Plastic Pipe and Fittings • Grinding Wheels • Packings • Tapes
Molded and Extruded Rubber and Plastic Products • Protective Linings and Coatings • Conductive Rubber • Adhesives • Roll Coverings • Mats and Matting

eliminate vibration effects. Developed by Meter and Instrument Dept., **General Electric Co.,** Schenectady 5, N. Y.

For more data circle MD-100, page 191

Continuous Printer

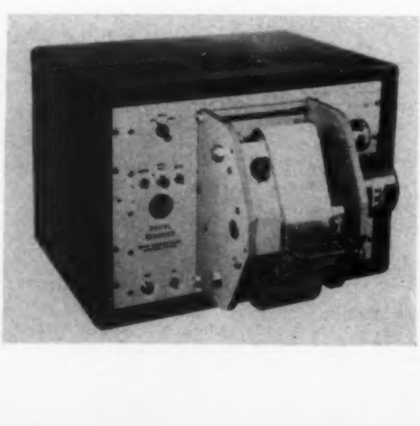
An automatic processing unit for making dry positive photo

copies. Copy is processed in 15 seconds and is ready for use in another 15 seconds. Unit eliminates washing, fixing, drying, and chemical fumes. Machine takes 12-inch paper in any length. Operates on 115 v ac and weighs 18 lb. Made of gray plastic, unit is available from **General Photo Products Co. Inc.,** General Photo Bldg., Chatham, N. J.

For more data circle MD-101, page 191

Digital Recorder

A new digital recorder, Model 960 announced by Potter Instrument, is for making high-speed, permanent digital recordings. This instrument, which will record up to 150 eight-digit numbers per second, has been designed for use with



counter - chronographs, frequency-time counters, scalers made by Potter or any similar equipment which provides binary coded decimal indication.

Data are recorded on electrosensitive paper by groups of four styli. Standard and special units are available with any number of channels up to 32. Paper speed is adjustable from 2.5 to 20 in. per second. Standard paper rolls 4¼ in. by 400 ft allow over 50,000 recordings to be made without reloading. Made by **Potter Instrument Co. Inc.,** 115 Cutter Mill Rd., Great Neck, N. Y.

For more data circle MD-102, page 191

Reproduction Camera

Capable of providing from five times reduction to two times enlargement, the Robertson "Seventeen" camera is precision engineered for simplicity of operation. A simple scaling system coupled with automatic timer permits foolproof operation. Maximum film

size is 17 by 17 in. and copyboard area is 21 by 25 in. Supplied with 14-inch focal length lens. Made by **Robertson Photo-mechanix Inc.,** 3067 N. Elston Ave., Chicago 18, Ill.

For more data circle MD-103, page 191

Frequency Meter

The Berkeley Model 5570 frequency meter is capable of measuring frequencies from 0 cps to 42 megacycles (160 megacycles with accessories). Accuracies of 1 part in 10^7 ± 1 count can be at-

tained. To obtain this high accuracy, provision has been made for externally adjusting crystal control unit to WWV. Indicated frequency is displayed in digital form on eight banks of illuminated

Lucite number panels. Instrument is 32 in. high, 21 in. wide, and 16 in. deep and requires a 117 v, 60 cps, 260 w power supply. Manufactured by Berkeley Scientific Div., **Beckman Instruments Inc.,** 2200 Wright Ave., Richmond, Calif.

. . **For more data circle MD-104, page 191**

Plan Files

For drafting and engineering offices, a new line of metal 5-drawer plan files have been announced. Useful for filing blueprints, industrial work plans and layouts, sketches and drawings, and any items that must be stored flat.

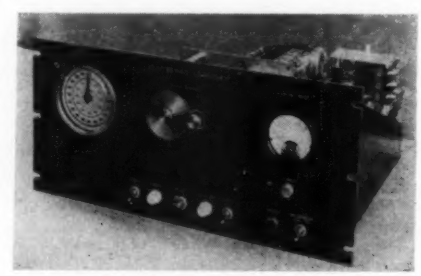
Drawers have ball bearing rollers and have positive locking feature which keeps them in position when in use. Finished in hammer gray enamel, the Mayline 5-drawer unit is constructed to allow stacking to whatever number desired. Made in many popular sizes by the **Engineering Mfg. Co.,** Sheboygan, Wis.

For more data circle MD-105, page 191

Counting Rate Computer

A new computer has been developed and designed to record random counting rates on a linear chart with a constant computed statistical accuracy. Unit utilizes a slidewire resistor driven by a synchronous motor which provides a circuit resistance proportional to time.

Computer must be used with a scaling circuit adjusted for fixed count operation. Modifications can be made to provide faster operation at low counting rates by use of the Cooke-Yarborough method but results are less accurate. Chassis layout provides for easy addition of accessory pulse generator. Developed by Research & Control Instruments Div., **North American Philips Co. Inc.,** 750 S. Fulton Ave., Mount Vernon, N. Y.



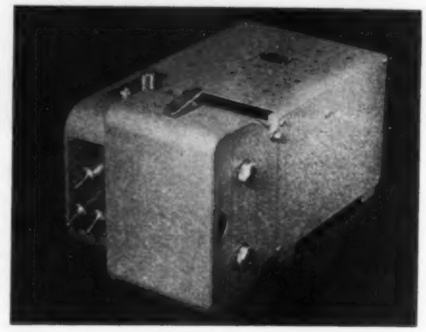
For more data circle MD-106, page 191

Multichannel Oscillograph

Among the important features of Midwestern's rugged new Model 580 oscillograph are 14 channels of separate data, full-width timing lines at 0.01-second with 0.1-second lines accentuated or 0.1-second lines only, and trace identification.

Engineering Equipment

Instrument incorporates a shutter-type "acceleration-proof" precision timer and has provision for record numbering. Full-width viewing screen displays galvanometer traces on one-to-one ratio with the traces being recorded; traces may be observed while oscillograph is operating. Precision galvanometers are



available with undamped natural frequencies up to 3500 cps.

Can be made to record at speeds of 1, 3, 6½, 13, 32, and 42 in. per second. Constructed of cast aluminum, unit is 7¼ in. wide, 6¼ in. high, and 12¾ in. long. Uses 3⅝ in. wide recording paper in lengths up to 100 ft. Manufactured by **Midwestern Geophysical Laboratory**, 3401 S. Harvard, Tulsa, Okla.

For more data circle MD-107, page 191

Strain Indicator

Operation with either batteries or ac power pack is the feature of the new model M Baldwin portable SR-4 strain indicator. An ac power pack is available that fits

Engineering Equipment

in the battery compartment and is mounted on the separate back cover of the instrument case.

Instrument provides direct readings in microinch per in. units up to 30,000 and can be used with an external Wheatstone bridge. Measuring accuracy is within ± 2.5 microinch per in. for the "thousands" step. Slide wire tolerance is ± 5 microinch per in. and gage factor adjustment is accurate within $\pm \frac{1}{4}$ per cent. Instrument is approximately 6 by 9 by 12 in. in size. Developed by **Baldwin-Lima-Hamilton Corp.,** Philadelphia 42, Pa.

For more data circle MD-108, page 191

Electronic Tachometer

Measurements of speed, frequency, or events per unit of time can be made with the new Standard electronic tachometer. Accuracy to ± 1 pulse is obtainable

depending on point in pulse generator cycle that the gate is opened and closed. Accuracy of time base is 1 part in a million. The instrument can be operated from any photoelectric, magnetic, electrical or electromechanical means producing pulses from 0.2 to 115 v without marked multiple peaks.

Circuit employs the use of 22 standard miniature tubes, 6 ten-element glow tubes, and 1 high-speed read-out unit containing 10 standard neon lamps. For easy servicing when required, instrument has an 11-unit plug-in subchassis. Made by **The Standard Electric Time Co.,** Springfield, Mass.

For more data circle MD-109, page 191


MICRO switches are "watchdogs" to prevent jamming or overload on this automatic labeling machine...
Three MICRO switches shown at feed table end of labeler provide automatic control of the label picker and gumming mechanisms
MICRO
MAKERS OF PRECISION SWITCHES
FREEPORT, ILLINOIS
A DIVISION OF
MINNEAPOLIS-HONEYWELL REGULATOR COMPANY

World Model B Automatic Labeler with accumulating tables for automatic infeed and discharge

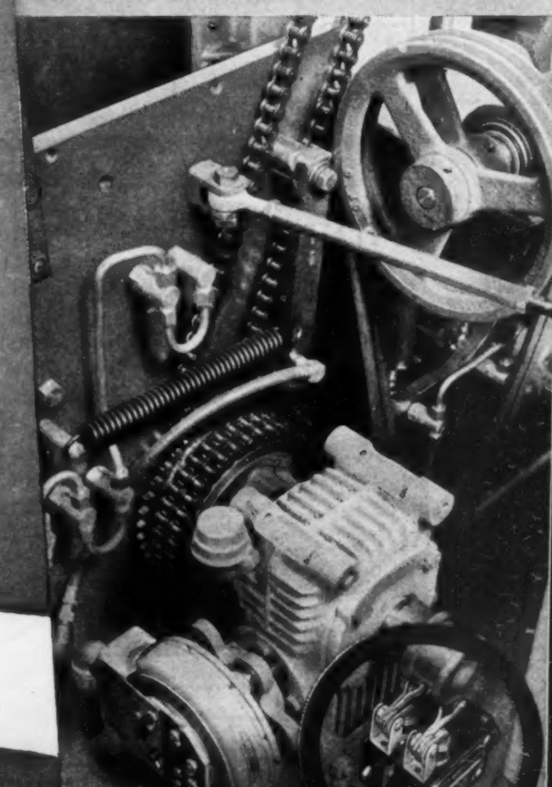
Five MICRO switches, located at strategic points, provide electrically operated "watchdogs" for this World Model B Automatic Labeler of the Economic Machinery Company, Division of George J. Meyers Manufacturing Company.

This ingenious machine is designed to label cylindrical vials or bottles at an adjustable speed range of from 90 to 180 bottles per minute. Automatic operation at such speeds depends on freedom from jamming because of free labels, broken glass or a bottle out of position.

Engineers of the Economic Machinery Company designed MICRO switches into the machine to serve as watchdogs at strategic points.

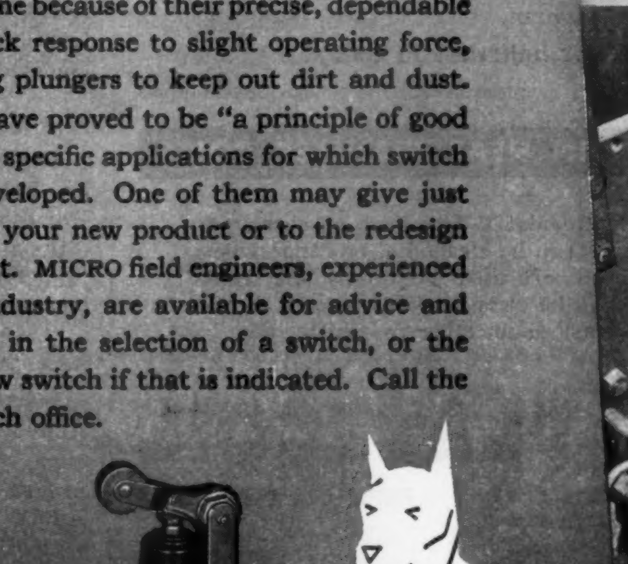
- **They prevent picking up a label unless there is a bottle in position to receive it.**
- **They stop the machine if a bottle is in position to be broken or to jam the machine.**
- **They control the label picking and gumming mechanisms.**

These two roller arm-actuated MICRO switches control the solenoid which prevents the machine from jamming or labeling when a bottle is not in position



MICRO switches were selected as components of this automatic labeling machine because of their precise, dependable operation, their quick response to slight operating force, and sealed actuating plungers to keep out dirt and dust.

MICRO switches have proved to be "a principle of good design" in over 6000 specific applications for which switch types have been developed. One of them may give just the "lift" needed to your new product or to the redesign of an existing product. MICRO field engineers, experienced in every phase of industry, are available for advice and counsel to help you in the selection of a switch, or the development of a new switch if that is indicated. Call the nearest MICRO branch office.



Meetings

AND EXPOSITIONS

Aug. 19-21—

Western Electronic Show and Convention to be held at Civic Auditorium, San Francisco, Calif. Additional information may be obtained from society headquarters, Merchandise Mart, San Francisco 3, Calif.

Sept. 6-11—

American Chemical Society. Fall meeting to be held at Hotel Conrad Hilton, Chicago, Ill. R. M. Warren, 1155 16th St., Washington, D. C., is assistant secretary.

Sept. 13-17—

Electrochemical Society Inc. Fall meeting to be held at the Ocean Terrace Hotel, Wrightsville Beach, N. C. Dr. Henry B. Linford, 235 West 102nd St., New York, N. Y. is secretary.

Sept. 20-23—

Packaging Machinery Manufacturers Institute. Twenty-first annual meeting to be held at the Skytop Lodge, Skytop, Pa. Additional information may be obtained from society headquarters, 342 Madison Ave., New York 17, N. Y.

Sept. 21-22—

Steel Founders' Society of America. Fall meeting to be held at the Homestead, Hot Springs, Va. Additional information may be obtained from society headquarters, 920 Midland Bldg., Cleveland, O.

Sept. 21-25—

Instrument Society of America. Annual meeting and joint conference with the Industrial Instruments and Regulators division of ASME to be held concurrently with the eighth national instrument exhibit at the Sherman Hotel, Chicago, Ill. Richard Rimbach, 921 Ridge Ave., Pittsburgh, Pa., is exhibit manager.

Sept. 28-30—

Association of Iron & Steel Engineers. Annual meeting to be held

Meetings and Expositions

at Hotel William Penn, Pittsburgh, Pa. T. J. Ess, 1010 Empire Bldg., Pittsburgh, Pa. is managing director.

Sept. 28-30—

National Electronics Conference. Ninth annual conference to be held at the Sherman Hotel, Chicago, Ill. under the sponsorship of the American Institute of Electrical Engineers, the Institute of Radio Engineers, Illinois Institute of Technology, Northwestern University, and the University of Illinois, with participation by Purdue University and the University of Wisconsin. S. R. Collis, N. E. C. Publicity Committee, 208 West Washington St., Chicago 6, Ill., is chairman.

Sept. 30-Oct. 2—

Procelain Enamel Institute. Twenty-second annual meeting to be held at the Greenbrier Hotel, White Sulphur Springs, W. Va. Additional information may be obtained from society headquarters, Dupont Circle Bldg., 1346 Connecticut Ave., Washington, D. C.

Oct. 5-7—

American Society of Mechanical Engineers. Fall meeting to be held at Hotel Sheraton, Rochester, N. Y. C. E. Davies, 29 West 39th St., New York 18, N. Y., is secretary.

Oct. 8-9—

National Conference on Industrial Hydraulics. Ninth annual meeting to be held at Hotel Sheraton, Chicago, Ill. J. G. Duba, Illinois Institute of Technology, Technology Center, Chicago 16, Ill., is secretary.

Oct. 12-13—

First Conference on Mechanisms. Two-day conference to be held at Purdue University, West Lafayette, Ind., under the joint sponsorship of the editors of MACHINE DESIGN and the faculty of the school of mechanical engineering at Purdue University. Additional information may be obtained from the Editor, MACHINE DESIGN, Penton Bldg., Cleveland 13, O.

DESIGN ABSTRACTS

DESIGNING
Worm Drives

A proposed method for standardizing general industrial coarse pitch cylindrical worm gearing

By F. G. East
Development Engineer
Hamilton Gear & Machine Co. Ltd.
Toronto, Ontario, Canada

WORM gearing, although one of the oldest types of gearing, is the one remaining type for which no recognized standard of design for the general industrial coarse pitch range exists on this continent.

There is an excellent ASME-ASA-AGMA standard covering the fine-pitch range but there is no standard in the general industrial coarse-pitch range. Lacking a standard, manufacturers of worm gears and worm gear reduction units have individually set up their own standards for tooth proportions, pressure angle, worm diameters, gear faces, etc. A standard recognized by the industry for all future designs is long overdue and would be of inestimable assistance to engineers and designers of new equipment.

Obviously if a limited number of gear sets are required, the cost of tooling is important. If design can be modified somewhat to suit existing facilities, it is advisable to consult one or more manufacturers to see what tooling is available that will produce gears close to the sizes desired rather than design strictly to a standard. However, the tentative design to submit to the manufacturer should be based on some recognized standard and for a new design that is to be made in sufficient volume to warrant the purchase of tooling, adherence to this standard is preferable and strongly recommended.

In developing standards for their own use, most manufacturers have

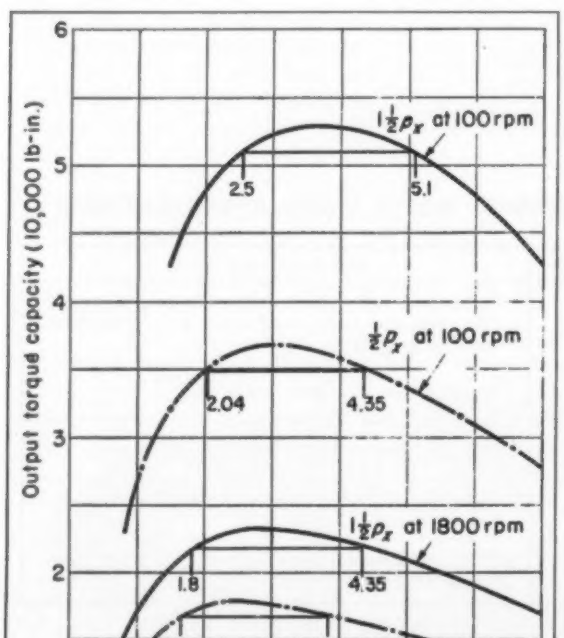
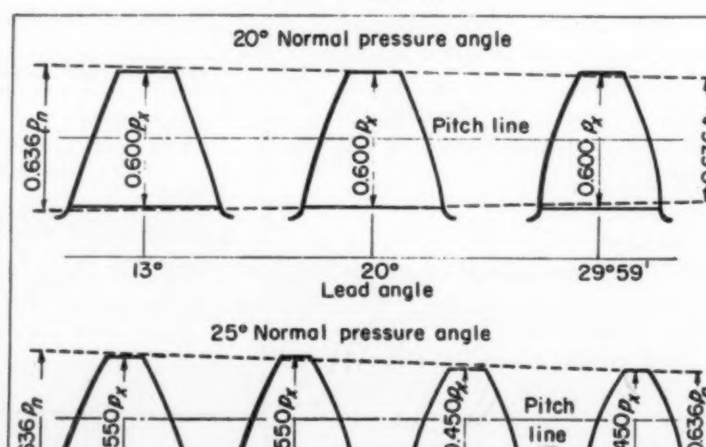


Fig. 1—Left—Power capacity of gear sets versus worm pitch diameters for various center distances and worm speeds

Fig. 2—Below—Cross-section of gear teeth for worms with various lead angles



found that good design usually limits its suitable pressure angles and tooth proportions for various lead angles. These are not calculated from the same bases by different manufacturers but the end results are very similar. The discussions and recommendations which follow apply to parallel cylindrical worms meshing with throated worm gears with the axes of the worms and gears at right angles. The worm proportions recommended may also be used for worms and gears meshing at angles other than 90 degrees.

The nominal pitch diameter of the worm (which for calculation purposes is assumed to be at the mean of the working depth) is the dimension most in need of standardization. Since no standard exists, no two designs will specify the same worm pitch diameter. Consequently since the hob which cuts the gear must match the worm diameter, a hob bought for one design will not be suitable for any other except by the most unlikely coincidence. Obviously the worm pitch diameter will, of necessity, be dependent on a number of factors including pitch, center distance, span between bearings, whether the worm is integral with its shaft or bored, the material in the worm, etc. Many of these factors are dictated by the design of other machine elements so that no one worm diameter for any given pitch will meet the requirements of all designs. However, this paper recommends a limited group of diameters for each pitch to eliminate the infinite number of diameters which will develop if no standard diameters are proposed.

Worm Pitch Diameters: If the power capacity of gear sets with worms of different pitch diameters is plotted for a given center distance, the maximum power capacity varies as shown in *Fig.* 1. The worm diameter that gives the maximum power capacity varies with the pitch and the worm speed. If a range of worm diameters that will give 95 per cent of maximum power capacity is considered good practice, the horizontal lines near the crests of the curves indicate that quite a range of worm diameters will satisfy this condition. Even if it is also assumed that most gear sets can run at different speeds, the range of diameters that will give power capacity within 95 per cent of maximum at all worm speeds between 100 and 1800 rpm will still be broader than required for most normal designs.

Based on the foregoing premise, graphs for different center distances and pitches similar to *Fig.* 1 were drawn. A number of points were calculated for the crown of the curves so this portion would be as accurate as possible. From these graphs, empirical formulas for maximum and minimum worm diameters were developed, TABLE 1. These are not absolute limits but

Table 1—Equations for Worm Dimensions

Lead

$$l = N_w p_x$$

Nominal Pitch Diam*

$$d_{max} = 0.53\ (0.745C + p_x)$$

or

$$C\left(\frac{3.33 + 0.395\,N_g}{1.67 + N_g}\right)$$

$$d_{min} = 0.51\ (0.388C + p_x)$$

or

$$C\left(\frac{3.20 + 0.198\,N_g}{1.6 + N_g}\right)$$

Outside Diam

$$d_o = d + 2a$$

Minimum Face

$$f = 2\sqrt{\left(\frac{D_t}{2}\right)^2 - \left(\frac{D_w}{2}\right)^2}$$

Lead Angle†

$$\lambda = \tan^{-1}\frac{l}{\pi d} = \tan^{-1}\frac{N_w}{\pi q}$$

*See recommended standard pitch diameters in TABLES 4 and 5. †See TABLE 4 for lead angles of standard worms.

Table 2—Equations for Gear Dimensions

Pitch Diam

$$D = \frac{N_g p_x}{\pi}$$

Throat Diam

$$D_t = 2C - d + 2a$$

Outside Diam*

Normal $D_o = D_t + a$

Maximum $D_o = D_t + 1.5\ a$

Effective Face Width

$$F_e = \sqrt{d_o^2 - d^2}$$

Helix Angle†

$$\psi = \tan^{-1}\frac{l}{\pi d} = \tan^{-1}\frac{N_w}{\pi q}$$

*Round off to nearest 1/16 inch. Maximum increment to be applied to low lead angle gearing only. †Same as lead angle of worm.

Table 3—Equations Common to Worm and Gear

Normal Pressure Angle	Lead Angle
$\phi = 20°$	$\lambda < 30°$
$\phi = 25°$	$30° \leqq \lambda \leqq 45°$
Addendum	**Lead Angle**
$a = 0.300\ p_x$	$\lambda < 30°$
$a = 0.275\ p_x$	$30° \leqq \lambda < 38°$
$a = 0.225\ p_x$	$38° \leqq \lambda \leqq 45°$
Whole Depth	**Lead Angle**
$h_t = 0.650\ p_x$	$\lambda < 30°$
$h_t = 0.600\ p_x$	$30° \leqq \lambda < 38°$
$h_t = 0.500\ p_x$	$38° \leqq \lambda \leqq 45°$
Working Depth	**Lead Angle**
$h_k = 0.600\ p_x$	$\lambda < 30°$
$h_k = 0.550\ p_x$	$30° \leqq \lambda < 38°$
$h_k = 0.450\ p_x$	$38° \leqq \lambda \leqq 45°$
Clearance	**Lead Angle**
$c = 0.050\ p_x$	All λ

Normal Pitch

$p_n = p_x \cos\lambda$
or
$= p_x \cos\psi$

Nominal Center Distance

$C = \frac{1}{2}\ (D + d)$

Nomenclature

C = Center distance, in.
D = Pitch diam of gear in central plane, in.
D_o = Outside diam of gear, in.
D_t = Throat diam of gear, in.
D_w = Working depth diam of gear in central plane, in.
F_e = Effective face width of gear, in.
N_g = Number of teeth in gear
N_w = Number of threads in worm
R_g = Gear gorge radius, in.
a = Addendum, in.
c = Clearance, in.
d = Nominal pitch diam of worm, in.
d_o = Outside diam of worm, in.
d_r = Root diam of worm, in.
f = Face width of worm (length), in.
h_k = Working depth, in.
h_t = Whole depth, in.
l = Lead, in.
p_n = Normal pitch of worm and gear, in.
p_x = Axial pitch of worm, in.
q = Worm pitch diam quotient
λ = Lead angle of worm, deg
ϕ = Normal pressure angle, deg
ψ = Helix angle of gear, deg

are recommended when applicable, because they keep the power capacity as high as possible.

Two formulas are given for both maximum and minimum diameters. One is based on center distance and pitch, and the other on center distance and number of teeth in the gear. In each case one is the algebraic transposition of the other, and the most convenient one can be used when designing a worm gear set.

Worms, which are to be integral with their shafts and have maximum efficiency, should lean toward the minimum diameter as calculated by the formulas, because the lead angle increases when the diameter is reduced (the higher lead angle produces higher efficiency). However, care must be taken to keep the root diameter of the worm large enough in relation to the span between the bearings to prevent undue deflection under load. Also on multiple-thread worms the diameter should be such that the lead angle does not exceed 45 degrees.

Worms, which are to be bored and mounted on their shafts will approach more closely the maximum diameter as calculated from the formulas. Bored worms should be kept as small as possible to obtain maximum lead angle and efficiency. However, they must have a root diameter large enough to allow suitable metal thickness between the bore or keyseat and the root of the thread.

Standardized Worm Pitch Diameters: It is recommended that the worm nominal pitch diameter be made *an integral number of half axial pitches* with multiples of full pitches preferred. This recommendation applies whether the design is within the minimum and maximum diameters obtained by the formulas or not. For example, the worm nominal pitch diameter should be 3, 3½, 4, 4½, etc., times the axial pitch with preferred worm diameters of 3, 4, 5, etc., times the axial pitch. The number of axial pitches in the worm pitch diameter is designated as the worm pitch diameter quotient, q. By using this system of fixing the worm nominal pitch diameter, a finite number of worm diameters is obtained for any range of sizes and makes possible the tabulation of worm and gear dimensions as shown in TABLES 4 and 5. It should be noted that worms of the same number of threads and worm pitch diameter quotient are geometrically similar and have the same lead angle regardless of pitch.

Pressure Angle: The pressure angle must be high enough to avoid undercutting on the leaving side of the worm gear teeth. As the lead angle increases, the minimum pressure angle that will prevent this undercutting also increases. The approximate maximum lead angles that can be used with various normal pressure angles are as follows:

Normal Pressure Angle (deg)	Maximum Lead Angle (deg)
14½	17
20	30
25	45

Although this table indicates that a 14½-degree normal pressure angle can be used for lead angles up to 17 degrees, it is recommended that its use be discouraged and 20-degree normal pressure angles be used for all worms with lead angles below 30 degrees and 25-degree normal pressure angles be used for worms with lead angles from 30 to 45 degrees. This is in line with

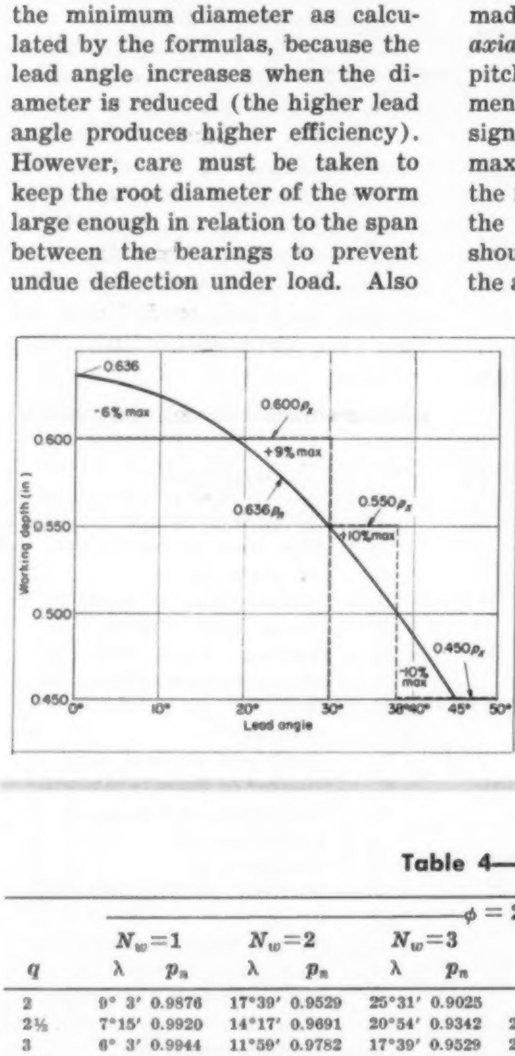


Fig. 3 — Variation of working depth values based on three values of axial pitch, p_x, from the smooth working depth curve developed from normal pitch, p_n for various lead angles

Table 4—Lead Angles and Normal Pitches*

	$\phi = 20°$												$\phi = 25°$					
	$N_w=1$		$N_w=2$		$N_w=3$		$N_w=4$		$N_w=6$		$N_w=8$		$N_w=4$		$N_w=6$		$N_w=8$	
q	λ	p_n	λ	p_n	λ	p_n	λ	p_n	λ	p_n	λ	p_n	λ	p_n	λ	p_n	λ	p_n
2	9° 3′	0.9876	17°39′	0.9529	25°31′	0.9025							32°29′	0.8436	43°41′	0.7232		
2½	7°15′	0.9920	14°17′	0.9691	20°54′	0.9342	26°59′	0.8911							37°23′	0.7946	45°32′	0.7005
3	6° 3′	0.9944	11°59′	0.9782	17°39′	0.9529	23° 0′	0.9205							32°29′	0.8436	40°20′	0.7623
3½	5°12′	0.9959	10°19′	0.9838	15°16′	0.9647	19°59′	0.9398	28° 37′	0.8778							36° 2′	0.8087
4	4°33′	0.9968	9° 3′	0.9876	13°26′	0.9726	17°39′	0.9529	25°31′	0.9025							32°29′	0.8436
4½	4° 3′	0.9975	8° 3′	0.9902	11°59′	0.9782	15°48′	0.9622	23° 0′	0.9205	29°30′	0.8704						
5	3°39′	0.9980	7°15′	0.9920	10°49′	0.9822	14°17′	0.9691	20°54′	0.9342	26°59′	0.8911						
5½	3°19′	0.9983	6°36′	0.9934	9°51′	0.9853	13° 2′	0.9742	19° 9′	0.9447	24°51′	0.9074						
6	3° 2′	0.9986	6° 3′	0.9944	9° 3′	0.9876	11°59′	0.9782	17°39′	0.9529	23° 0′	0.9205						
6½	2°48′	0.9988	5°36′	0.9952	8°21′	0.9894	11° 5′	0.9814	16°22′	0.9545	21°24′	0.9311						
7	2°36′	0.9990	5°12′	0.9959	7°46′	0.9908	10°19′	0.9838	15°16′	0.9647	19°59′	0.9398						
7½	2°26′	0.9991	4°51′	0.9964	7°15′	0.9920	9°38′	0.9859	14°17′	0.9691	18°45′	0.9469						
8	2°17′	0.9992	4°33′	0.9968	6°48′	0.9930	9° 3′	0.9876	13°26′	0.9726	17°39′	0.9529						

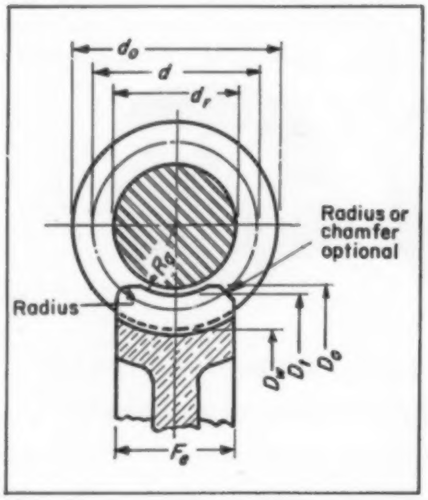
*Dimensions shown in inches for axial pitch of 1 inch.

Design Abstracts

current practice to discontinue the use of the 14½-degree pressure angle as a standard for gears of other types and also conforms with the fine-pitch standard.

Pitch: The pitch should be specified parallel to the axis of the worm. This is known as the axial pitch of the worm and is the same as the circular or transverse pitch of the gear. This is common practice and is most satisfactory due to the necessity of gearing up worm milling and grinding machines to the worm lead. It is suggested that wherever possible one of the following pitches be used. These pitches are considered close enough together to satisfy most new designs. When intermediate pitches are necessary, they should be midway between the recommended pitches, provided this is possible. The recommended axial pitches are as follows: 3/16, ¼, 5/16, ⅜, ½, ⅝, ¾, 1, 1¼, 1½, 1¾, 2, 2¼, 2½, 2¾, and 3 inches.

Fig. 4 — Gear meshing with worm showing critical dimensions



Tooth Proportions: The tooth proportions (addendum working depth and whole depth) have to be reduced in relation to the axial pitch as the lead angle increases to compensate for reducing the normal pitch.

It is common practice to make the tooth or thread depth of standard full depth tooth proportions based on the axial pitch for the lower lead angles and the normal pitch for the higher lead angles. Since the pitch is specified in the worm axial plane for all lead angles, it is also logical to proportion the tooth depths from the axial pitch for all lead angles. In *Fig.* 2, which shows normal sections of worms of various lead angles, the broken lines above and below the pitch line that converge to the right show standard full depth tooth proportions based on the normal pitch for worms with lead angles which are shown along the base line. The full lines show teeth proportioned from the axial pitch for each group of lead angles.

In *Fig.* 3 it is shown that the maximum variation of working depth using the recommended proportions of the axial pitch varies from the full-depth proportions based on the normal pitch by not more than 10 per cent. This variation is negligible compared to the advantage of having only three tooth depths for each pitch over the whole range of lead angles from 0 to 45 degrees instead of an almost infinite number when based on the normal pitch. It is also more logical, as mentioned previously, to proportion the tooth depths from the pitch in the plane in which it is specified, the same as is done in other types of gearing.

Thread Form: The thread form of the worm must match that of the hob used to cut the gear. Worms ground with straight-sided V-wheels or milled with similar shaped milling cutters will have a form that will vary with the wheel or cutter diameter. This variation is much greater on worms with high lead angles than on those with low-lead angle worms. Therefore, it is important that the method of production and the shape of the grinding wheel or milling cutter used to produce the worm be taken into consideration in determining the required profile of the hob teeth. It is preferable to have both worm and gear made by the same manufacturer to insure matching tooth shapes and proper conjugate tooth action.

Thread & Tooth Thickness: The normal gear tooth thickness should never be less than half the normal pitch with the backlash allowance

Table 5—Worm and Gear Dimensions*

		$\lambda = 30°$				$30° \leqq \lambda < 38°$				$38° \leqq \lambda \leqq 45°$			
		$\phi = 20°$				$\phi = 25°$				$\phi = 25°$			
		$a=0.300$, $h_k=0.600$, $h_t=0.650$				$a=0.275$, $h_k=0.550$, $h_t=0.600$				$a=0.225$, $h_k=0.450$, $h_t=0.500$			
q	d	d_o	d_r	F_e	R_g	d_o	d_r	F_e	R_g	d_o	d_r	F_e	R_g
2	2.000	2.600	1.300	1.661	0.700	2.550	1.350	1.581	0.725	2,450	1.450	1.414	0.775
2½	2.500	3.100	1.800	1.833	0.950	3.050	1.850	1.746	0.975	2.950	1.950	1.565	1.025
3	3.000	3.600	2.300	1.990	1.200	3.550	2.350	1.897	1.225	3.450	2.450	1.703	1.275
3½	3.500	4.100	2.800	2.135	1.450	4.050	2.850	2.037	1.475	3.950	2.950	1.830	1.525
4	4.000	4.600	3.300	2.272	1.700	4.550	3.350	2.168	1.725	4.450	3.450	1.949	1.775
4½	4.500	5.100	3.800	2.400	1.950	5.050	3.850	2.291	1.975	4.950	3.950	2.062	2.025
5	5.000	5.600	4.300	2.522	2.200	5.550	4.350	2.408	2.225	5.450	4.450	2.168	2.275
5½	5.500	6.100	4.800	2.638	2.450	6.050	4.850	2.520	2.475	5.950	4.950	2.269	2.525
6	6.000	6.600	5.300	2.750	2.700	6.550	5.350	2.627	2.725	6.450	5.450	2.366	2.775
6½	6.500	7.100	5.800	2.857	2.950	7.050	5.850	2.729	2.975	6.950	5.950	2.460	3.025
7	7.000	7.600	6.300	2.960	3.200	7.550	6.350	2.828	3.225	7.450	6.450	2.550	3.275
7½	7.500	8.100	6.800	3.059	3.450	8.050	6.850	2.924	3.475	7.950	6.950	2.636	3.525
8	8.000	8.600	7.300	3.156	3.700	8.550	7.350	3.017	3.725	8.450	7.450	2.720	3.775

*Dimensions shown in inches for axial pitch of 1 inch.

(if any) taken off the worm. In some special cases advantage can be made of the usual difference in strengths of the worm and gear material by making the thickness of the gear teeth more than half the pitch. When this is done, however, consideration must be given to the tip land on the worm. For general practice this is usually not necessary.

Number of Threads: Wherever possible, the number of threads (or starts) in the worm should be limited to one of the following—1, 2, 3, 4, 6, or 8. This simplifies indexing on worm grinding and milling machines. The use of 5 or 7 threads is discouraged. If a greater number of threads than 8 are required, numbers which are not odd or prime should be used to facilitate production of the worm without special indexing equipment.

Number of Teeth in Gear: The permissible minimum number of teeth in the gear is influenced to a considerable extent by the center distance. On the smaller center distances, adherence to too large a minimum (which might be preferable from a tooth action standpoint) would result in a pitch too small. In the larger center distances, it is preferable not to make the pitch too coarse which dictates the minimum number of teeth to a great extent. The following recommended minimum number of teeth for various center distances are considered good general practice:

Center Distance (in.)	Number of Teeth (min)
2	20
3	25
5	25
10	29
15	35
20	40
24	45

Gear Ratio: The gear ratio is equal to the number of teeth in the gear divided by the number of threads in the worm. Opinions differ on whether the number of teeth in the gear should be prime to or an integral multiple of the number of threads in the worm. In practice either prime or even ratios generally give equally satis-

factory results. Suppose a hob is not available to cut a gear which is to mesh with a multithread worm and a single tooth "fly cutter" is to be used. In this case it is unnecessary to "index" the gear or cutter on the hobbing machine if the number of threads in the worm is prime to the number of teeth in the gear.

Center Distance: The center distance is normally half the sum of the pitch diameters of the worm and gear. It is not essential that the pitch circle of the worm gear be at the mean of the working depth. In fact when there are sufficient teeth in the gear and the pressure angle is high enough to prevent undercutting, the pitch line can be anywhere between the mean of the depth and the throat diameter of the gear or even outside the throat. This produces a short addendum on the gear and lengthens the angle of recess to improve the lubrication characteristics and load carrying capacity.

For similar reasons the pitch circle preferably should not be inside the mean of the working depth because this produces a long gear tooth addendum, lengthens the angle of approach, worsens the lubrication characteristics, and tends to reduce the load carrying capacity. In other words, worm gears should preferably not be operated on centers larger than half the sum of the nominal pitch diameters of the worm and gear. Operation on less than standard centers allows simple fractional center distances to be used.

When the center distance is made less than one-half the sum of the pitch diameter of the gear and the nominal pitch diameter of the worm, the actual operating pitch diameter of the worm is less than the nominal pitch diameter or the diameter at the mean of the working depth. In such a case the operating pitch diameter of the worm is equal to twice the center distance minus the pitch diameter of the gear. (The pitch diameter of the gear is always equal to the number of teeth times the circular pitch divided by pi). This does not affect the worm calculations or

Design Abstracts

production which is based on the nominal pitch diameter.

Unlike standard involute spur and helical gears, the form of worm gear teeth varies across the face. If the worm thread form is straight sided on the axial section, the worm gear teeth will be involute in the central plane. Regardless of the worm tooth form, in off-center planes the gear teeth are not involute and are only conjugate to the mating worm when operated on or near the center distance for which they were cut. The allowable tolerance between cutting and operating centers is much less with worms of high lead angle than with those of low lead angle.

Worm Face: The worm face or length should extend beyond the points where contact between it and the teeth of the worm gear begins and ends. Since the pressure angle of a worm gear varies across the face and becomes quite low on the leaving side, contact between it and the worm extends almost to the point where the outside diameter of the worm intersects the throat diameter of the gear. The face of the worm should be made equal to twice the distance from the line of centers to this intersection point to be safe. The formula for f in TABLE 1 fulfills this requirement.

Gear Face: The effective worm gear face varies with the nominal pitch diameter of the worm and the depth of thread. The formula for effective gear face, F_e, given in TABLE 2 is based on the maximum effective face of the worm gear which is the length of a tangent to the mean worm diameter between the points where it is intersected by the outside diameter of the worm. Any additional face has little value and wastes material. The sharp corners at the point where the gear face intersects the outside diameter should be removed by the use of a radius or chamfer as shown in *Fig.* 4. The size of this radius or chamfer is not critical but should be approximately equal to 0.25 times the axial pitch.

Gear Outside Diameter: The gear outside diameter is not critical and is usually made approximately equal to the throat diameter plus one addendum (0.5 addendum added to the throat radius). The resultant figure is then rounded off to the nearest 1/16-inch. On low lead angle (slow-speed gearing) this increment can be increased to one and a half addendums added to the throat diameter but this should never be exceeded. This latter amount should not be applied to gears with helix angles exceeding 15 degrees.

Tabulated Data: The nomenclature gives suggested symbols for worm and gear dimensions. Based on the foregoing, the author suggests the formulas in TABLE 1 for worm dimensions, in TABLE 2 for gear dimensions, and in TABLE 3 for dimensions applying to both worms and gears. These would be a good basis for a worm-gear standard that would conform to the best modern practice. For convenience TABLE 4 lists the lead angles, normal pitches, and recommended pressure angles based on 1-inch axial pitch for worms with diameter quotients from 2 to 8 and recommended number of threads ranging from 1 to 8. The normal pitch of worms of other axial pitches with the same diameter quotient and number of threads are in direct proportion, and the lead angles are the same regardless of pitch.

TABLE 5 gives worm and gear data for worms of 1-inch axial pitch and various diameter quotients. Similar data for worms of other pitches are in direct proportion.

If such a system of worm diameters based on diameter quotients were adopted in a design standard, a complete list of sixteen tables giving these data for all the recommended pitches could be included as part of the standard. This would simplify the calculations for the design of worm gear drives which conformed to the standard and would encourage its use.

From a paper entitled "A Proposed Standard Design for General Industrial Coarse Pitch Cylindrical Worm Gearing" presented

Design Abstracts

at the ASME Spring Meeting in Columbus, O., April 1953.

The Automatic Factory

By D. S. Harder and D. J. Davis

Ford Motor Co.
Dearborn, Mich.

IN discussing automatic handling in factories, there are several major considerations which should be pointed out to provide a clear understanding of the problems involved. Following is a list of the important subjects:

1. Transfer type machinery built with automation in mind. The transfer type machine combines similar operations to be performed on a part into one continuous machine with automatic indexing between work stations.
2. "Automation"—a coined word meaning the automatic handling of parts between progressive production processes. It is a separate and distinct phase of manufacturing engineering which co-ordinates the efforts of various departments in plants and of machinery and equipment manufacturers.
3. Preventive maintenance programs to keep production lines in operation, including tool changing and standardization.
4. A reorganization and reallocation of work tasks and manpower to place a greater emphasis on brain power instead of muscle power.
5. Alert management to co-ordinate and get the best out of each man's technical skills.

Transfer Type Machinery: For most production lines of a few years ago, processing of manufactured parts was based on a series of individual machine tools performing the operations in small increments with manual handling between each machine. Processing today must be based on the combination of similar operations into an integrated machine tool of many stations with automatic transfers between.

To illustrate the techniques used, consider the production of automotive crankshafts. One portion of the machining process involves the drilling of six oil holes, six meter-

ing holes, six lightening holes, and inspection of the operation. Ten years ago 29 separate machines would have been required, while today the same department requires only three transfer type machines of eight stations each. The part is loaded at one end and unloaded at the other with all operations in between being completely automatic. This example also applies to other operations such as milling, broaching, reaming, and tapping.

Obviously combinations of this type cause profound changes on entire production processes. Manpower must be redistributed to handle the problems which arise. A fresh look must be taken at methods of maintaining machinery elements and in applying tools to the job. Production lines in many cases are found made up with groups of transfer machines which are completely automatic in themselves and which in turn must be coupled together in an economical fashion if full benefits are to be gained.

Automation: While similar operations are combined with the transfer machine, there is a separate problem still remaining of connecting these machines so that manual handling operations will not be necessary.

Automation devices are designed to move parts into and out of production machines, turn the part over, rotate it, remove scrap, and other related functions. Scrap removal on a continuous and automatic basis is a necessity in both machining and stamping operations. These devices are timed with the process and are ordinarily made up largely of standard elements such as conveyors and air, hydraulic, and electrical control mechanisms arranged to obtain the proper movement for elimination of unnecessary manual handling. A wide variety of materials can be handled with substantial savings, while at the same time eliminating the hazardous handling associated with large stampings, heat treat operations, forging operations, and others. In many respects automation is the extended usage of conveyors which have played such

"Not A Flight of Fancy"

HERE ARE WAYS CHIKSAN CAN HELP AIRFRAME MANUFACTURERS

The airplane shown here is not an actual picture of any one plane. Rather, it is a composite to show some of the varied ways in which Chiksan planning can help the airframe manufacturer do a better job.

Fuel, oil, water, air and oxygen all find dependable, safe, uninterrupted flow through Chiksan aero-hydraulic swivel joints and assemblies.

CHECK THESE APPLICATIONS IN YOUR BLUEPRINTS:

- **A.** Refueling boom assembly
- **B.** Steering control mechanism
- **C.** Brake system
- **D.** Spoiler actuating mechanism
- **E.** Fuel and oil lines
- **F.** Adjustable sweep-back system
- **G.** Emergency dump chute
- **H.** Flap actuating system
- **I.** Power-plant fuel and oil lines
- **J.** Wing fold
- **K.** De-icing system
- **L.** Power control system
- **M.** Stabilizer trim

The Flow of Enterprise Relies on

CHIKSAN

Ball-Bearing Swivel Joints

REPRESENTATIVES IN PRINCIPAL CITIES

A few of the many Chiksan aero-hydraulic swivel joints made for pressures ranging from 1000 to 3000 psi.

an important part in the early development of mass production of automobiles.

Production Costs: Lower production costs are an important consideration in all facilities planning. Automation assists greatly by relief of manual effort required in handling parts and reduction of the resultant inherent delay, especially in large volume operations. This does not mean elimination of all manpower, but rather a redistribution which will result in the need for more skilled workmen. It eliminates monotonous, repetitive work in many cases while at the same time increasing production.

Increased Machine Speed: The speed at which a machine produces a part is very often determined by the loading and unloading operations. Automatic transfer between operations has resulted in greater use factors for machines. The machinery has been able to operate closer to its designed capacity. Using high cost machinery to a greater degree is, of course, a definite advantage.

Maintenance Costs: Use of automation devices in the production processes naturally causes maintenance problems which could offset the savings to be made if not properly handled. As a production line becomes more automatic breakdown of even the smallest element can shut down the entire line. This maintenance problem can be minimized as skill in automation design is increased. The details of systems must be made as simple as possible and devices must be made substantial. Preventive maintenance is a must.

Nature of Automation Equipment: Every product requires special consideration. In the manufacture of a wide variety of parts of different sizes and shapes, the machinery and production equipment utilized varies in size. Proper emphasis on automation makes it possible to study the problems at the beginning of new programs so that standardized automation devices can be used to minimize this difficulty.

An automation designer must have imagination, engineering knowledge, and design experience. He must think in terms of equipment which very often is a compromise between the comparatively loose standards used for designing a conveyor and the close standards utilized in normal tool and die design. This is the type of man who starts analyzing a program in the first step when the general plant layout has been agreed upon and detailed layouts are in their early stage.

A part to be produced must be carefully studied for adaptability to automation. Preliminary manpower estimates must be considered. One operator may be able to load and unload several machines, or may be required for assembly, manual control, observation, or others; automation may not even be necessary.

Preventive Maintenance: Maintenance of automatic machines and automation equipment is a challenge to resourcefulness. As pointed out previously, the breakdown of even one small element will shut down the entire production line. Preventive maintenance is the key to continuing production and involves generally the keeping of accurate records of lubrication, expected life of perishable parts, and replacing these parts before they fail and stop production. One very effective tool which can be used is standardization of tools, methods and specifications, industrial equipment, process standards, and materials standards. These standards are based on exhaustive surveys of alternate methods before they are accepted. Preventive maintenance, like all other features of a modern production plant, begins in the early stages of planning and takes advantage of the new devices which are available to ease the maintenance burden.

Future of Automatic Processes: The use of automation in our manufacturing plants is a trend which has started and will continue and expand. The future will certainly find a much greater use of the knowledge which has been obtained within the past few years. Increased management, attention engineering thought, and co-ordination will keep this trend going and

Design Abstracts

extend it substantially for maximum benefit to all.

The future of automatic processes is inseparable from the future of our manufacturing industries in the constant fight to reduce cost so that consumption of our manufactured goods may be increased, and our employment may remain at a high level.

From a paper of the same title presented at the National Production meeting of the SAE in Cleveland, O., March 1953.

Spinning or Drawing?

By John W. Lengbridge

Project Engineer
Aluminum Goods Ltd.
Newcastle-on-Tyne, England

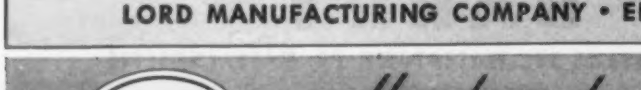
SPINNING may be described as a method of working a revolving disk to shape over a form or chuck to which it is clamped, causing the metal to flow by tooling pressure applied either manually or mechanically. Unlike drawing, in which the change of shape is hidden by the tooling, shaping by spinning is visible throughout the operation.

Drawing metal to shape consists of working it into a hollow die of predetermined shape by means of a punch, the movement of metal being controlled with a blankholder which prevents the metal from buckling.

Limitations of Shapes: Shaping by pressing is not limited to any one specific shape; it can be used for circular as well as noncircular products within the capacity of the presses used. By contrast, spinning is limited to products which are circular in cross sections at right angles to the axis of rotation, except for some minor work which can be done on ovals. Within its field the scope of spinning is wide, and it is limited only by the size of the equipment and the power available to make the metal flow. An evaluation of the two processes must be confined to circular products which can be made on either type of equipment. It will be shown that spinning can be competitive as

(*Continued on Page* 270)

The new Super-Dome Passenger Cars of the Milwaukee Railroad are air conditioned to maintain comfortable temperature at all times. A 20 ton capacity Trane Compressor and a 20 ton capacity Trane Condenser in each car do this important job. Lord Mountings protect these Trane Units from vibration and shock and prevent transmission of the unit vibration to the car thus assuring passenger comfort. In these ultra-modern cars the passengers enjoy the benefits of healthful, temperate air. This is another of the many examples of Lord versatility in assisting designing engineers to solve difficult mounting problems. You are invited to consult with us on the application of Lord Vibration and Shock Mountings and Bonded-Rubber parts to improve the operation of your product.



BURBANK, CALIFORNIA 233 South Third Street	**DALLAS, TEXAS** 413 Fidelity Union Life Building	**PHILADELPHIA 7, PENNSYLVANIA** 725 Widener Building	**DAYTON 2, OHIO** 410 West First Street
DETROIT 2, MICHIGAN 311 Curtis Building	**NEW YORK 16, NEW YORK** 280 Madison Avenue	**CHICAGO 11, ILLINOIS** 520 N. Michigan Ave.	**ERIE, PENNSYLVANIA** 1635 West 12th Street

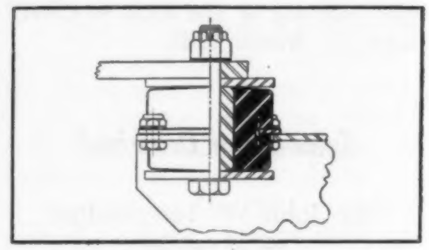
LORD MANUFACTURING COMPANY • ERIE, PA.

LORD BONDED RUBBER *Headquarters for* **VIBRATION CONTROL**

Advertisement

Railroad Equipment Protected By Lord Mountings

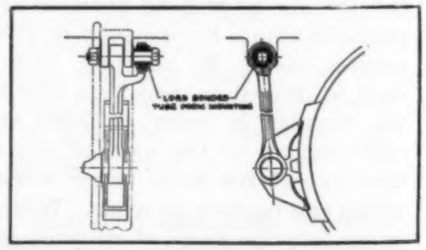
A special plate form mounting has been designed by Lord Engineers for use as a shock mounting for engine-generator and motor-compressor units installed on railroad rolling stock. The problem was to design a Lord Mounting which would minimize the shocks of coupling and uncoupling cars and thus prevent damaging the undercar equipment referred to above.



An existing Lord plate form mounting design was utilized with a steel tubing insert in the elastomer to increase the stiffness of the mounting. Tooling and research expense were thus saved by this unusual application of Lord Mountings previously designed to solve another similar problem.

Excessive Wear Eliminated By Use of Lord Mountings

A recently designed Lord Mounting has solved a problem for a manufacturer of brake assemblies in the railroad field. A Lord Tube Form Mounting design was utilized as a joint for the brake hanger mechanism. Through the use of rubber bonded to metal sufficient motion was permitted to compensate for shock while eliminating wear on the pin and joint at the point of motion.

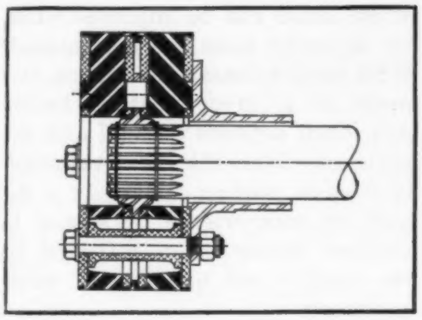


Lord Engineers determined the amount of resiliency required to effect a firm joint while permitting sufficient movement to eliminate excessive wear. An existing design was used with the correct elastomeric compound to meet the necessary requirements. Thus a considerable saving was achieved.

Advertisement

Lord Flexible Coupling Flies With Kaman Helicopter

The Kaman K-5 helicopter presents an interesting application of the LORD J-5329-2 flexible coupling with the Boeing 502-2 gas turbine. The function of the coupling in this case is to absorb the torsional vibrations of the system and isolate the turbine from the rotors. However, the unit also serves to accommodate angular or parallel misalignments due to manufacturing tolerances or dynamic motions.



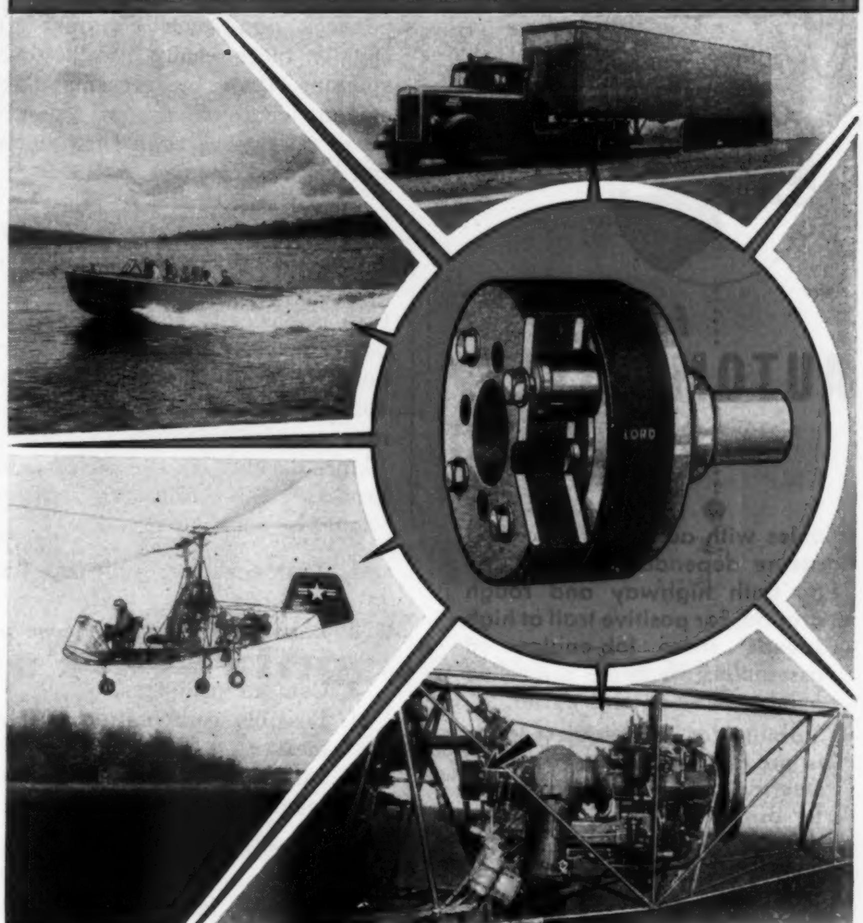
The unique design of the installation provides maximum accessibility and economical maintenance through the use of concentric driving and driven shafts. The inner member of this pair is the engine shaft which drives the coupling hub through a splined connection. Precompressed against the splined hub are the two bonded rubber coupling halves which transmit the engine torque in shear of the rubber. Four through-bolts connect the outer plates of the coupling halves to the driven hub and also serve as the safety interlock in case the rubber sections are destroyed. The first gear of the transmission is mounted on this driven hub and feeds power on through the system in to the helicopter rotors.

For over thirty years the Lord organization has specialized in designing and producing Bonded Rubber Flexible Couplings, Vibration and Shock Control Mountings and Component Parts. The capabilities of Lord Engineering have proved their worth to designers of industrial and automotive equipment in many diversified fields as is indicated in this instance.

ON LAND, SEA AND AIR

LORD *Flexible Couplings*

TRANSMIT THE FULL POWER OF BOEING TURBINE ENGINES



HERE again you see at a glance Lord versatility in designing bonded-rubber components for a wide diversity of machines. The photo at top right shows the Boeing Gas Turbine-Driven Truck-Trailer for heavy cargo hauling. At the top left you see a United States Navy personnel boat driven by the Boeing Gas Turbine Engine. Directly beneath is the Kaman Helicopter powered by the Boeing Gas Turbine Engine; details are clear in the foreground. The Lord Bonded-Rubber Flexible Coupling designed for the job transmits the power in each machine.

Special requirements like these reach satisfactory and economical solutions at Lord, Headquarters for Vibration Control. We invite you to take advantage of more than a quarter century of design experience and craftsmanship.

BURBANK, CALIFORNIA 233 South Third Street	**DALLAS, TEXAS** 413 Fidelity Union Life Building	**PHILADELPHIA 7, PENNSYLVANIA** 725 Widener Building	**DAYTON 2, OHIO** 410 West First Street
DETROIT 2, MICHIGAN 311 Curtis Building	**NEW YORK 16, NEW YORK** 280 Madison Avenue	**CHICAGO 11, ILLINOIS** 520 N. Michigan Ave.	**CLEVELAND 15, OHIO** Room 811 Hanna Building

LORD MANUFACTURING COMPANY • ERIE, PA.

LORD BONDED RUBBER *Headquarters for* **VIBRATION CONTROL**

Design Abstracts

(*Continued from Page* 267)

well as supplementary to press drawing and that each process is logical under certain sets of circumstances.

Re-entrant contours are a frequent requirement in domestic appliances and such contours are usually more easily produced by spinning than by pressing. Re-entrant contours can be spun in fewer operations than such shapes can be press formed.

Time and Cost Criteria: The decision to use either spinning or drawing as a method of manufacture is basically economic and depends largely on the quantity required. When the equipment for both processes is available, spinning is an alternative to drawing. Before deciding on the use of either process the following factors should be considered:

1. Tool cost per unit by both methods.
2. Labor and overhead cost per unit.
3. Contour problems.
4. Possible combination of both methods.
5. Dimensional accuracy requirements.

Spinning is predominantly a manual operation and as such usually involves high labor cost. The nature of the operation, however, permits the use of simple and economical tooling. Often it can be mechanized, and although this may increase tool and equipment costs, it brings unit labor costs closer to those of drawing.

Drawing is a mechanical operation capable of low labor costs, but requires more costly tooling. These components of the cost factor must be analyzed in relation to lot size before a comparable cost can be arrived at.

Tool costs vary with quantity while labor cost is more or less constant. Hence, the total unit cost including tools is largely dependent on the cost of the tools necessary to do the work. Since the variable in both cases is the cost of tools, which in spinning is a low component and in drawing a high component. it follows that the unit cost of a drawn product would drop more rapidly with an increase in quantity than it would with a spun product. However, up to a certain point in the quantity scale the high cost of press tools may bring the unit cost above that of spinning. It is this demarcation point which must be determined when analyzing the probable cost of drawing or spinning.

Accuracy Requirements: Greater accuracy of size and wall thickness and better uniformity are possible on drawn work than on spun work. On a hand spun product, a tolerance of plus or minus 1/32-inch is as close as can be guaranteed. These limits can be improved when the spinning tools are mechanized. With hand spinning the human element is a predominating factor and much depends on skill and experience. Wall thickness variation in a spun product is largely a result of inexperience. Fatigue is another factor often reflected in the quality and quantity of work produced.

Capacity Limitations: The press is not limited to any one specific shape as is the spinning lathe. However, it is limited to a certain die size, maximum drawn depth, and tonnage capacity, whereas a spinning lathe can if necessary be altered to swing a larger blank by merely jacking up the head and tail stock. It can also be fitted with supplementary equipment for special operations or for boosting the spinning pressure.

Competitive Aspects: Spinning can be competitive to drawing on small-lot runs because of low tooling costs. Spinning is often competitive also because the contour of a shell is either difficult or impossible to produce on a press. Occasionally a shell which is too large for available press equipment can be spun to shape economically, and there are certain products which regardless of quantity can be more economically spun than drawn. Special attachments to standard spinning lathes to mechanize tool application and special spinning machines for unusual operations demonstrate progress in spinning techniques which tend to close the gap between spinning and drawing.

The question naturally arises,

What he's proving about pumps 15 MILES UP...

may hold an idea YOU can use

A shaft seal, for example, that stays vapor-tight even at —70°F. And at an atmospheric pressure only 1/30 that at sea-level.

Or pumps to handle low-boiling-point liquids, (like aviation gasoline), without troublesome vapor-lock or cavitation.

Or pumps to work at 1,000-pound pressure dependably in desert heat *and* Arctic cold.

* * * * *

These, and many other pumping problems have been solved by the Accessories Division . . . then *proved* in our High-altitude Lab. where engineers can "fly" a pump to 75,000 feet without ever leaving the ground.

Your pump requirements, present or future, can find a prompt solution from the broad experience the Accessories Division has had in aircraft fuel pump development. After all, Thompson Products is the leader in manufacturing pumps for aircraft and commercial production. Write us, describing your application.

ACCESSORIES DIVISION

Thompson Products, Inc.

DEPT. AM-7 • CLEVELAND 17, OHIO

Design Abstracts

"What metals can be spun?" Spinnability is directly proportionate to ductility, and a metal which can be deep drawn can usually be deep spun. This fact may be used as a guide in selecting metals for spinning. The high formability possible by spinning results from the progressive nature of the operation which is equivalent to many successive drawing operations. Metals which are difficult to draw may often be spun, but may require more power and possibly the use of heat during spinning or else anneals between operations.

Spinnability: In order to tabulate the property of spinnability, the Milwaukee Metal Spinning Company has conducted tests on various metals and arrived at a spinnability rating for several of the more commonly used metals. The ratings shown in Table 1 are based on a numerical value of 1.00 for aluminum alloy 2SO which in the tests indicated maximum spinnability. Some of the metals which

Table 1—Spinnability of Metals

Metal	Shallow Spinning	Deep Spinning
Aluminum and Magnesium		
Aluminum 2SO and 3SO	1.00	1.00
Aluminum 57SO and 55SO	0.98	0.62
Aluminum 17SO and 24SO	0.65	0.45
Magnesium	0.55	0.45*
Miscellaneous Soft Metals		
Zinc	0.94	0.94
Pewter	0.94	0.93
Lead	0.90	0.85
Steels		
Cold rolled (deep drawing steel)	0.91	0.91
Hot rolled (pickled drawing steel)	0.91	0.72
Lead coated (long ternes)	0.90	...†
Galvanized	0.82	...†
High tensile steel	0.40	0.13
High carbon (0.40% and up)	0.22	0.09
Copper and Copper Alloys		
Commercial bronze	0.87	0.87
Copper (cold rolled annealed)	0.87	0.87
Yellow stamping brass	0.86	0.86
Nickel silver (up to 30%)	0.85	0.75
Phosphor bronze (soft temper)	0.73	0.39
Muntz metal	0.47	0.13
Nickel and Nickel Alloys		
Nickel	0.86	0.86
Monel (deep drawing quality)	0.86	0.82
Inconel	0.81	0.75
Stainless Steels		
347 (18/8)	0.67	0.67
430 (14/18)	0.67	0.53
304 (18/8)	0.65	0.65
302 (18/8)	0.65	0.33

* Special set up for hot spinning. † Coating will flake.

have good deep-drawing properties are also high in spinnability; others which are difficult to draw have fair spinnability.

Considering drawn or spun shells having either parallel-sided or tapered walls, it may be stated as a general rule that contours which provide the best control of flow in a drawing operation do not provide as good control in spinning operations, and conversely contours providing poor control in drawing operations provide better flow control in spinning operations.

As mentioned previously, mechanization of some or all of the hand spinning technique leads the way to lower spinning cost. Thick or tough metals require pressures far in excess of human brawn. For such applications, semimechanical or fully mechanical means must be used to apply the pressure necessary to make the metal flow.

In this evaluation, it must be recognized that each process, spinning and drawing, is logical under certain circumstances. Each is capable of the complete manufacture of products within its scope, and both can be used on some products to good cost advantage. Hence they can compete as well as supplement each other, and whether one or both processes are used depends mainly on the economics of each individual case.

From a paper entitled "An Evaluation of Spinning vs. Drawing" presented at the Twenty-First Annual Meeting of the ASTE in Detroit, Mich., March, 1953.

Applying Teflon in Design

By L. W. Cornell

Development Engineer
Minnesota Mining & Mfg. Co.
St. Paul, Minn.

TEFLON is the trade name for a comparatively new polymeric resin with unique properties which is composed of tetrafluoroethylene, C_2F_4. This material has already been utilized in a variety of applications, and it offers a wide range of design possibilities. Here in a single material is a combination of toughness, abrasion resistance, chemical resistance, excellent elec-

Design Abstracts

trical properties and high and low temperature resistance.

Electrical Properties: Teflon is a versatile electrical insulator for many purposes, particularly where high temperature conditions must be met. It may be used at temperatures as high as 500 F in continuous service and withstands intermittent exposure to 600 F. Also, flexibility and electrical properties are retained to temperatures as low as minus 100 F. It has a uniform dielectric constant and power factor over a wide frequency range and the broad temperature range enables it to be used in applications where other less expensive materials would not be suitable. There are many uses for Teflon insulated wire and cable in jet aircraft, for example. Guided missiles are another application where the plastic is being used because of the temperature range encountered.

Aircraft motors and generators can be made smaller with Teflon as slot liners, etc. Coils and relays which must operate continuously at high temperatures use the material for interlayer insulation and for providing insulated magnet wire in the coils themselves.

High temperature resistant capacitors are now being manufactured with Teflon sheet insulation instead of paper or other material. The plastic is also being molded into tube sockets for radio and radar components for aircraft and into insulating spacers for coaxial cable.

Chemical Properties: Chemically Teflon is about as inert as glass within its useful temperature range. It is attacked by molten alkali metals (sodium or potassium) and by fluorine and chlorine trifluoride, both at high temperatures and pressures. Other than these specific chemicals, published literature indicates that nothing else has any effect on this plastic. This makes it useful as a gasket material and liner in equipment in which high temperature, highly corrosive chemicals must be handled. Since there is no known solvent for this material, it can be used in contact with any ordinary solvent without effect. Outside weathering has no effect and samples exposed to natural aging for 5 years in Florida have shown no change.

In large stainless-steel or glass-lined equipment many gaskets are needed. The main body of the gaskets used on such equipment is generally made of asbestos, lead, rubber, cloth, or a combination of these materials in order to provide the desired physical characteristics of compressibility, etc. These materials must be protected from the action of the chemicals inside the equipment, and this can be done by the use of a Teflon shield which is shaped to cover the top and bottom faces and the inside edge. Such gasket shields vary in size from a little over 1 in. ID to 5 ft ID.

This plastic is also expected to find application as liners or bladders in connection with containers for white and red fuming nitric acid, when certain fabrication problems have been solved. It has the ability to give chemical protection in thin flexible sheets and has withstood nitric and sulfuric acid mixtures as gaskets on equipment handling these acids at about 400F. Teflon gaskets gave 18 months service in comparison with one month for asbestos.

Molded tubing with fairly thick walls is available for chemical work and large pipe with relatively thin, tough walls composed of a combination of glass cloth and Teflon is now being produced. Small diameter tubing with very low wall thickness (as low as 0.005-in.) can also be manufactured.

Mechanical Properties: Its mechanical properties are stable up to 500 F. Molded bars held at 480 F for one month have shown a loss of only 1 per cent in tensile strength. At 570 F the loss in tensile strength was 10 to 20 per cent. It is serviceable to −100 F; uses at temperatures as low as −320 F have been reported.

The nonadhesiveness or slipperiness of this material has proved to be useful in the industrial field. For example, one large coating machine was giving trouble with dirtying up of idler rolls in the oven on a certain job which was performed at intervals. The problem was solved by spiral wrapping wide Teflon film around the roll, taping it down at each end of the roll. When the operation was finished, the film was unwound and

stored until it was needed again.

Certain parts of pumps and valves used in highly corrosive service are being fabricated from Teflon. It can be molded, ground,

Table 1—Properties of Teflon

Property	Value
Tensile strength, 77 F.	1500-2500 psi
Elongation, 77 F	100-200 %
Stiffness, 77 F	60,000 psi
Impact strength (Izod), −70 F	2.0 ft-lb/in.
77 F	4.0 ft-lb/in.
170 F	6.0 ft-lb/in.
Hardness-Durometer ..	D55 to D70
Compressive stress, 0.1% deformation...	1700 psi
Coefficient of linear thermal expansion, 77 to 140 F	5.5×10^{-5}/deg F
Thermal conductivity..	1.7 Btu/hr/sq ft/deg F/in.
Specific heat	0.25 Btu/lb/deg F
Deformation under load, 122 F for 85 hrs at 1200 psi	4 to 8%
Heat distortion temperature, 66 psi	270 F
Water absorption	0%
Specific gravity	2.1 to 2.3

and machined. A detailed list of properties is given in TABLE 1.

Forms Available: The value of Teflon to the designer stems from the fact that it is provided in many forms. A multiplicity of uses have resulted and many new uses are being made of the material.

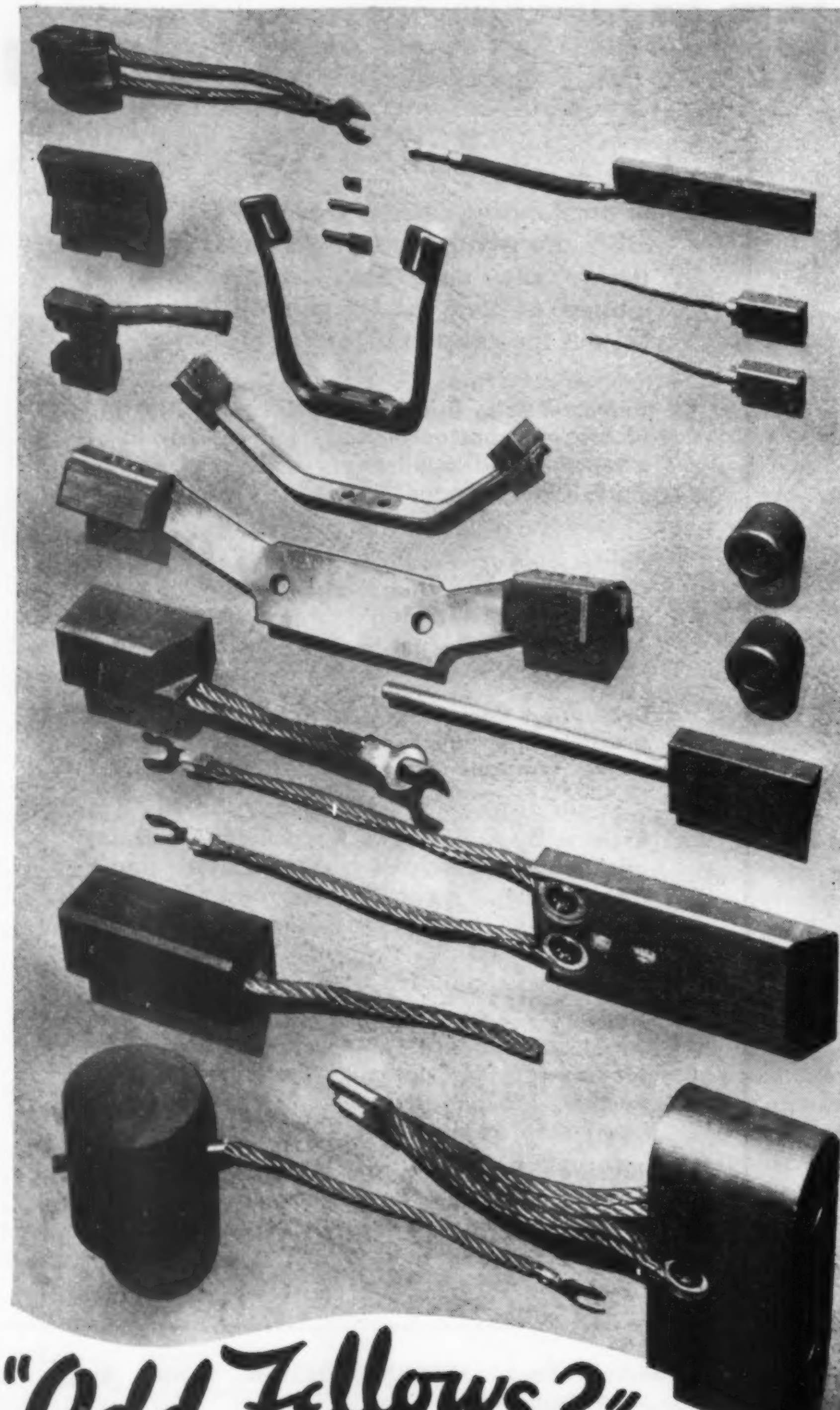
MOLDING POWDER: This is one of the basic forms made. It is granular and is used for making heavy sheet, molded shapes, etc.

MOLDED PIECES: Sheets 1/16-in. thick or heavier as well as heavy-walled tubing, rods, bars, special shapes, etc. can be produced. In this grouping there should be included molded gaskets, valve packings, valve seats, etc. At least one company manufactures molded Teflon with copper or other metal firmly bonded to one face. Because of its high heat resistance, solder can be applied directly to the metal sheath.

SUSPENSOID: This is another basic form, but its particle size is much smaller than in molding powder. It is used for casting films, roll coating, spray coating, etc.

WIRE ENAMELS: These are suspensoids specially processed for use in wire coating for the electrical industry.

METAL PRIMER: Three primers, one for copper, one for aluminum, and one for iron or other metals,



are manufactured. They give improved anchorage of Teflon coatings to these metals.

LUBRICATED PASTE: A specially prepared form, this paste can be extruded for certain applications and is made from coagulated dispersion.

Films

SHAVED FILM: Film in various calipers is shaved from the circumference of a log or bar of the plastic. This shaved film is satisfactory for many applications where low pinhole count is not important. It is not oriented or tensilized to any extent.

EXTRUDED FILM: Made from the lubricated paste, extruded film is highly oriented and possesses greatly increased tensile strength in the machine direction. Two forms of extruded film are made.

Transparent Extruded Film: Produced as a film that is fully fused, it is used in a wide variety of applications including motor slot liners, cable wrappings, capacitor dielectrics, coil wrappings, gasket shielding for corrosive chemicals, etc. It is a transparent, smooth surfaced film with low pinhole count. Dielectric strength runs as high as 2500 to 3000 volts per mil in 2 to 5-mil caliper film.

Self-fusing Extruded Film: Made in white and opaque forms, this product will stretch and is conformable. It is seldom used in its unfused form, mainly because it is too soft and can be cut or abraded too easily. Its value lies in the fact that it does not slide too easily before fusion and that it bonds firmly to itself when fused in contact. It can be wrapped around articles which must be electrically insulated, or which must be protected from chemicals, and then fused in place to give a tough continuous covering. Because of its softness and pliability, it conforms well to various shapes around which it is wrapped. Of course, this sheet form can be fused only onto metal or other material which can stand the high fusing temperature.

CAST FILM: Cast in multiples, this film is made from suspensoid and is not oriented or tensilized.

PRESSURE SENSITIVE TAPE: Pres-

Spots Usually Spell Trouble

While that's true of little boys *and* castings, the condition is not equally obvious. In the case of castings, spots or flaws are usually unseen . . . invisible to normal inspection . . . but present nonetheless and ready to cause trouble.

That's why the highest standards of control and exacting inspection accompany a Sivyer casting through every step of its manufacture. This constant vigilance is your best assurance of flawless, precision steel castings.

The sign of a "healthy" casting. The Sivyer (S) is your guarantee of a safe, dependable casting — always look for it.

SIVYER

SPECIALISTS IN HIGH ALLOY AND SPECIFICATION STEEL CASTINGS

SIVYER STEEL CASTING COMPANY • MILWAUKEE (S) CHICAGO (S) MAIN OFFICE: 1675 SO. 43rd ST. • MILWAUKEE, WIS.

Design Abstracts

sure-sensitive, Teflon film-backed tapes have been made in combination with glass cloth. The adhesive used is one which vaporizes more or less completely at the maximum temperature to which the Teflon might be subjected in service, leaving little or no carbon residue. The tape simplifies the job of holding the plastic material in place during fabrication.

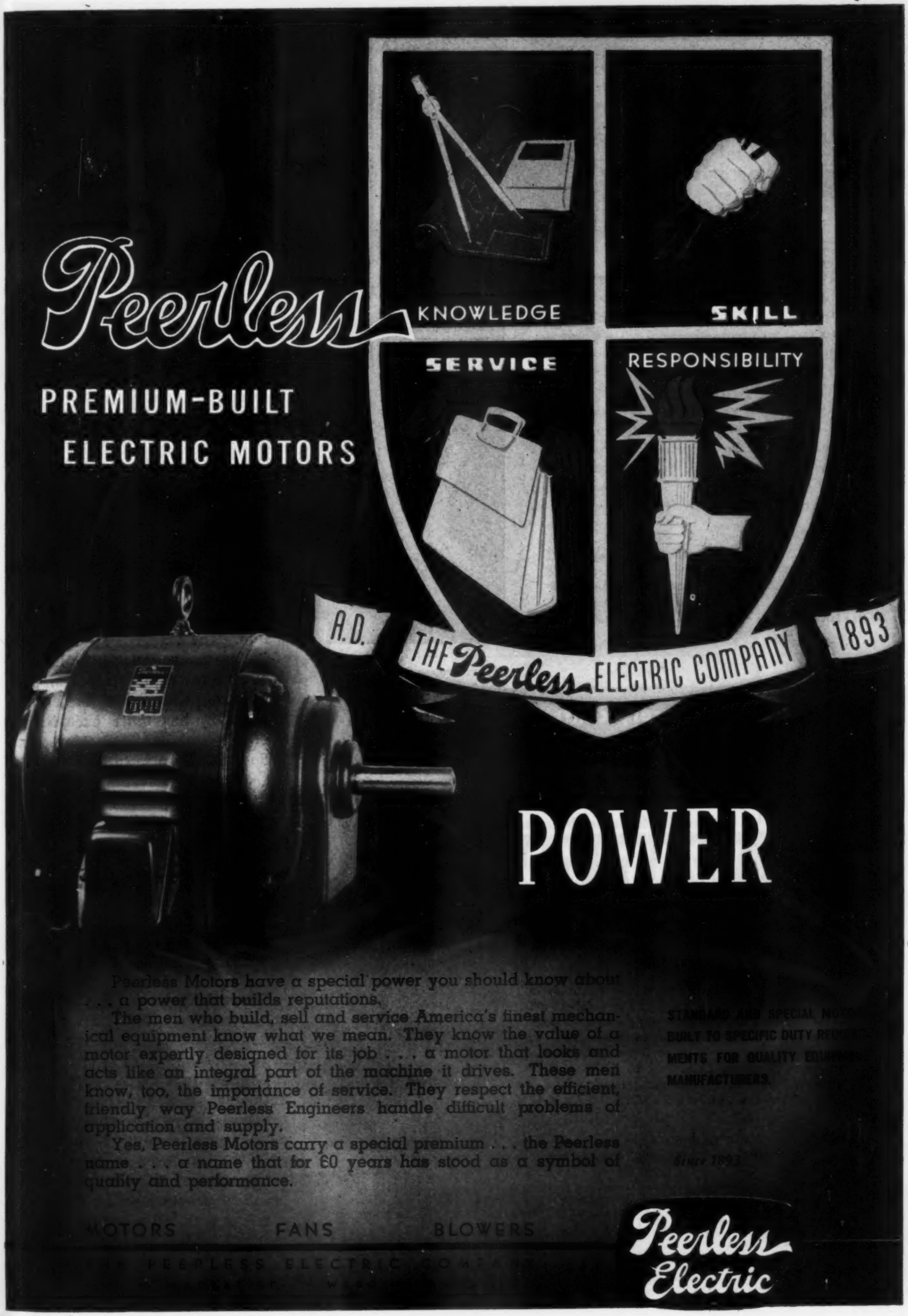
Methods of Fabrication: In order to develop the inherent toughness and strength of Teflon, it must be fused or sintered above its transition temperature of 621 F. At this temperature the plastic changes from a fairly hard, semitransparent, waxy solid to a rather soft, amorphous, transparent gel. This change occurs at 621 F whether the plastic has previously been sintered or not. Before this sintering is done the first time, the plastic is not suitable for most operations.

The various types of dispersions may be applied by spray gun or by dipping. They must then be dried and sintered at about 750 F. Successive coats are required to eliminate pinholes and give good chemical protection, and each coat should be separately fused.

Tapes can be used as a spiral wrap, which is frequently done for electrical insulation. For chemical protection, however, the self-adhering type which must be sintered after wrapping offers a better means of securing sealed protection.

The white self-fusing film sheet is easy to apply and to fuse around small shapes of regular surface, the simplest being wire. Wrapped wire can be fused continuously by passing through a suitable oven maintained at 750 to 850 F or higher if the wire is quite small. The fused wire should be quenched in water as soon as it comes from the oven. The film may be applied over a wide variety of sizes and shapes, but special care is frequently required to insure fusion without crack formation.

Toxicology: A recent technical information bulletin on Teflon from du Pont states the following: "Minute amounts of gaseous fluorine compounds are given off at tem-


Peerless
PREMIUM-BUILT
ELECTRIC MOTORS
KNOWLEDGE
SKILL
SERVICE
RESPONSIBILITY
A.D.
THE Peerless ELECTRIC COMPANY
1893
POWER
Peerless Motors have a special power you should know about . . . a power that builds reputations.
The men who build, sell and service America's finest mechanical equipment know what we mean. They know the value of a motor expertly designed for its job . . . a motor that looks and acts like an integral part of the machine it drives. These men know, too, the importance of service. They respect the efficient, friendly way Peerless Engineers handle difficult problems of application and supply.
Yes, Peerless Motors carry a special premium . . . the Peerless name . . . a name that for 60 years has stood as a symbol of quality and performance.
Since 1893
MOTORS
FANS
BLOWERS
Peerless Electric

peratures above 390 F and appreciable quantities are given off at 600 F or above. At approximately 750 F polytetrafluorethylene decomposes slowly. Adequate ventilation must be provided for any operation where these temperatures may be attained, because of possible toxicity of these gases."

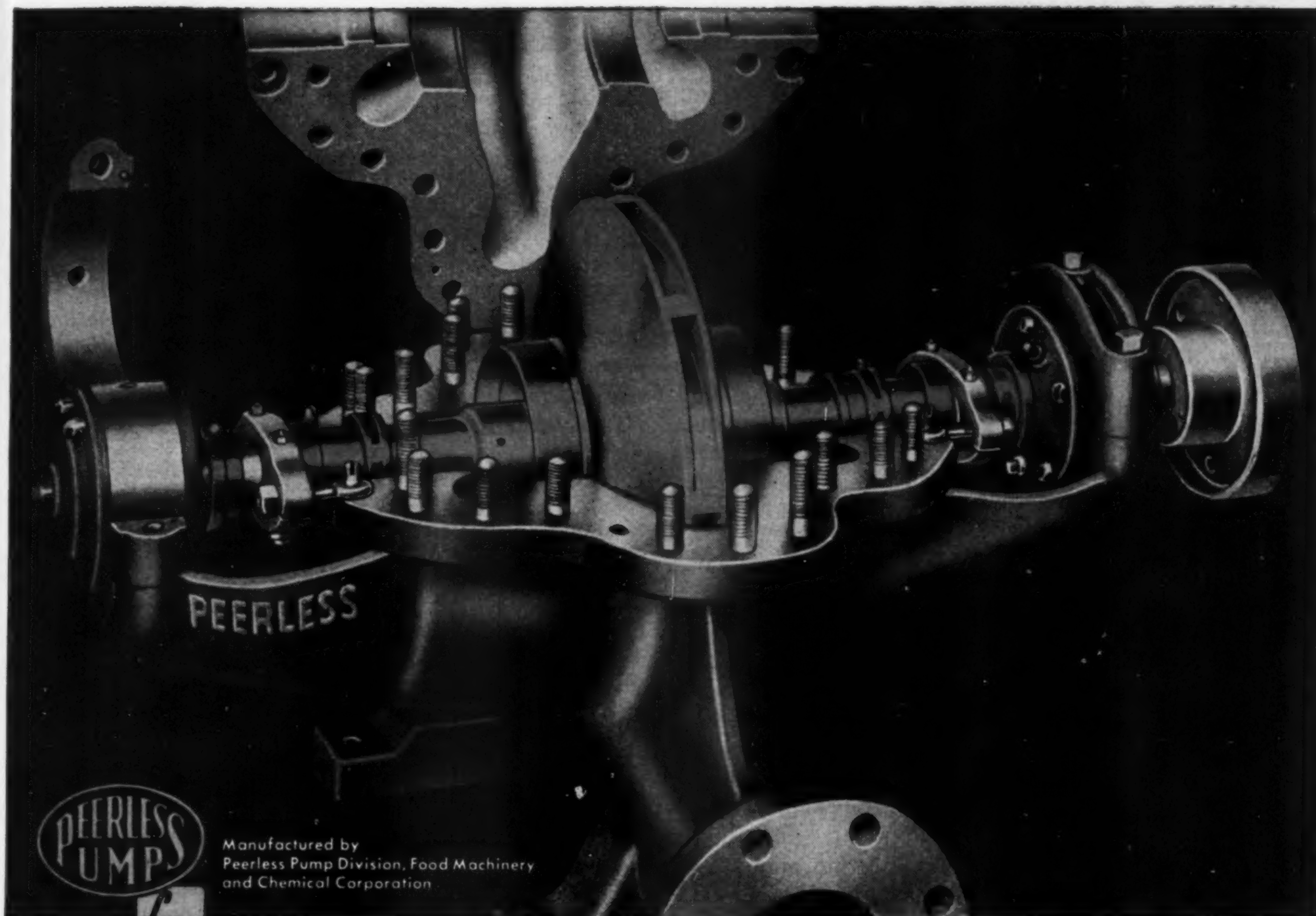
From a paper entitled "Polytetrafluorethylene—Its Properties and Uses" presented at the Spring Meeting of the ASME in Columbus, O., April 1953.

Lubricating with Molybdenum Disulphide

By Alfred Sonntag
President
The Alpha Corp.
Greenwich, Conn.

MOLYBDENUM disulphide, MOS_2, which resembles graphite in appearance, is a mineral and is the primary source for molybdenum. The basis for the lubricity of molybdenum disulphide lies in its unique molecular structure. By comparison with other materials, pure MOS_2 has quite exceptional chemical stability and inertness. It does not dissolve in cold or boiling water, in solvents, petroleum oils, or synthetic lubricants. It resists the attacks of most acids except aqua regia, boiling concentrated hydrochloric acid, fluorine and chlorine. In the presence of pure oxygen it oxidizes at room temperature. Since the sulphur is in chemical combination, it does not react on metals like free sulphur.

Thermal Stability: The lubricating properties of MOS_2 are not affected by subzero temperatures and in the presence of air it has practical applications up to 750 F. Oxidation in air at atmospheric temperatures is so slow that it takes many years to notice it, and then only the top layer exposed to the air is affected. Oxidation reaches practical proportions at 750 F increasing with rising temperature to a very rapid decomposition at 1100 F. The product of decomposition in air is molybdenum trioxide which has no lubricating proper-



When fire strikes IT PUMPS PROTECTION

This is the heart of reliable systems that protect against hazards of fire

America's annual loss to the ravages of fire is a staggering amount. Reliable plant and property protection often begins with an independent fire protection system, the heart of which is an approved and reliable fire pump. In the manufacture of fire pumps, rigid construction requirements and operational dependability are paramount. This is why National Bearing Division was selected to supply bronze impellers, wearing rings, and shaft sleeves, for this fire pump and other pumps bearing the Peerless name.

In supplying component fire pump parts to Peerless, a supplier must not only meet the rigid requirements of this manufacturer, but also the strict regulations of Underwriters' Laboratories, Inc.; National Board of Fire Underwriters, Chicago; and the Inspection Department, Associated Factory Mutual Fire Insurance Companies, Boston.

National Bearing's proved ability to mass produce exceptionally fine-grained, nonferrous castings, free from blow holes and sand inclusions, combined with Peerless Pump Division's exacting engineering and manufacturing standards help build fire pumps, and industrial pumps that can be counted to pump a reliable water supply for fire protection or other plant needs.

If your product requires non-ferrous castings or bearings, National Bearing Division has the foundry facilities, experience and skill that will insure better product performance and may possibly lower product costs.

AMERICAN **Brake Shoe** COMPANY

NATIONAL BEARING DIVISION

4931 Manchester Avenue • St. Louis 10, Mo.

PLANTS IN: ST. LOUIS, MO. • MEADVILLE, PA. • NILES, OHIO • PORTSMOUTH, VA. • ST. PAUL, MINN. • CHICAGO, ILL.

ties, and, depending upon particle size, may act as an abrasive. In inert atmospheres, decomposition temperatures are considerably higher—1600 to 2000 F in a vacuum and 2400 to 2800 F in argon have been reported.

Methods of Application: Aside from MoS_2 in fine powder form, a variety of MoS_2 grease consistency mixtures, colloidal and near-colloidal suspensions in petroleum oil, synthetic lubricating liquids, or solvents is now available.

There appears to be much promise in developing methods of bonding coatings without the use of binders. Rubbing the powder into a bearing surface by hand applies a coating which serves many purposes, but the intimate contact necessary to bond is only made at the hilltops of the surface. Hard mechanical rubbing, pressing and impacting by tumbling makes for a more uniform and effective bond. Such mechanical forces pound or squeeze the material for intimate contact with the metal into the grooves and force out vapor layers and pockets. There are indications that the employment of heat may contribute to better bonding of the lubricant.

Applying highly concentrated homogeneous MoS_2 grease consistency mixtures to bearing surfaces and running them in for short periods usually produces effective and antigalling low-friction coatings. After the running-in period, excess material may be removed by solvents or other conventional degreasing methods, if dry lubrication is desired. Friction tests prove that repeated thorough cleaning with all type of solvents will not remove a basic coating of MoS_2. Abrasive methods or a cleaning procedure which includes pickling in muriatic acid have been used to remove all traces.

Design Properties: Natural and pure MoS_2 is a solid extreme pressure lubricant whose effectiveness in comparison with other known lubricants increases with bearing pressure, providing lubrication for pressures in excess of 400,000 psi, far beyond the yield point of any metal. Its function as a lubricant is independent of temperature over a range of at least −90 F to 750 F, and for even higher temperatures in the absence of air.

Experience indicates affinity for bonding to metal surfaces. Ultimate effectiveness appears to depend on methods used to obtain a uniform and solid bond with bearing surface. The aim is a surface layer of molybdenum disulphide molecules which are not separated from each other or from the bearing surface by liquid or gaseous films.

Service

As a dry lubricant it is serving successfully

1. Where liquid films cannot be maintained for reasons of heat or cold.
2. In dusty atmospheres where liquid films are undesirable.
3. Where the thickness of liquid films interferes with the accuracy of measurement on optical and other measuring equipment for example. It has been possible to work with zero or even negative clearance between surfaces without indications of galling or seizing.
4. In cases where the deterioration of liquid films into gummy residues ultimately impairs the proper function of certain mechanisms.

MoS_2 or its lubricating compounds are successfully used where extreme bearing pressures cause galling (scuffing), seizing, welding, and excessive wear of bearing surfaces; for reciprocating parts where liquid lubricants are difficult to maintain and fretting results; for lubrication of mating surfaces made of similar metals—stainless steel on stainless steel, etc.; as an EP additive to other lubricants, including silicones and other synthetic materials lacking EP properties; for lubrication of natural and synthetic rubber; and as an additive to cutting oils or to cold-forming coolants.

From a paper entitled "Properties and Uses of Pure Molybdenum Disulphide as a Lubricant" presented at the meeting of the American Society of Lubrication Engineers in Milwaukee, Wis., March **1953.**

New Machines

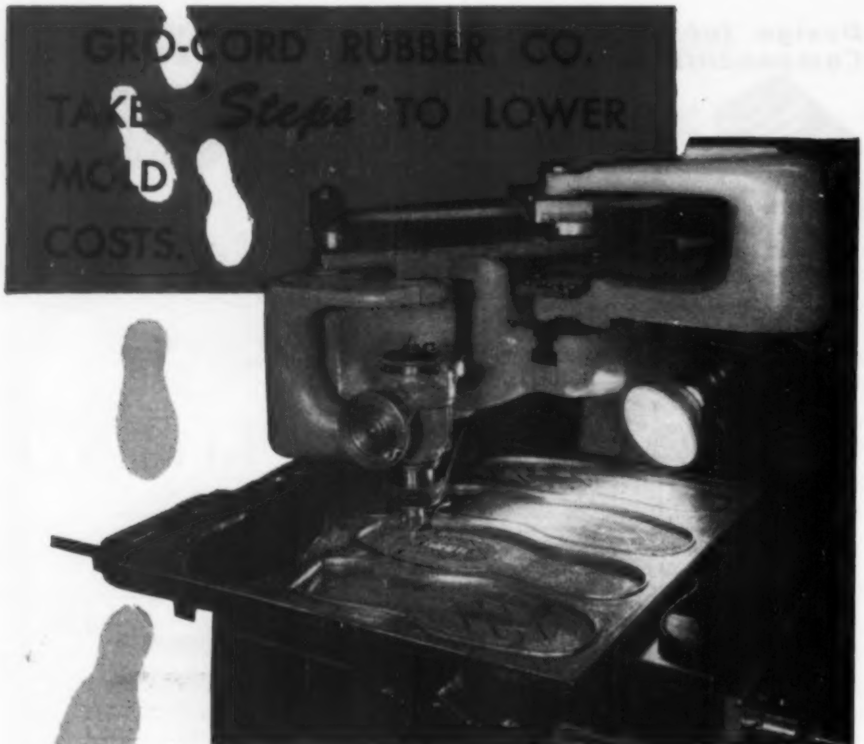
Materials Handling

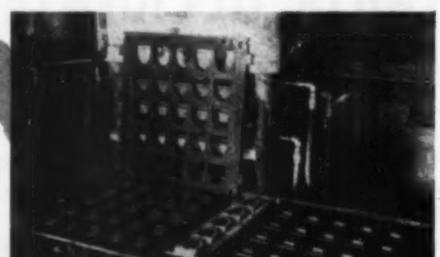
Chain Conveyors: Available in 3½ and 4½-in. widths; handle bottles, cans, lumber and work in process. Constructed of hot roll steel or stainless steel; various drives, speeds and chain design meet requirements of installation and operation. Clean-out holes along both top sides provide easy accessibility to underside of chain links and inside of frame. Completely self-contained; power drive is mounted at one end of frame. Sides of flat, hinge type chain ride on a smooth channel. Guide rails available. *Sage Equipment Co., Buffalo, N. Y.*

Feeding Table: Portable; has load capacity of 3 to 10 tons. For feeding large, heavy sheets to presses, brakes or shears. Top measures 36 by 96 in.; is hydraulically operated by multiple-speed foot pump, and is also available with electric drives. *Rack Hydraulic Equipment Corp., Connellsville, Pa.*

Sheet and Strip Handler: Coilveyor supports long, narrow strips as they are cut by a shear. Can handle strips up to 6 ft long and from 20 to 48 in. wide. After ribbon comes off coil, passes through leveler, it is cut by the shear. Air-operated plungers of this machine support the ribbon as it butts against a stop acting as a back gage. As shear cut is completed, cut ribbon, or strip, is lowered to conveyor belts and carried away. Plungers return automatically to support next cut. *Fried Steel Equipment Mfg. Corp., New York, N. Y.*

Electric Tiering Truck: Reach-Fork has 2000-lb capacity. Reaches out to pick up or deposit loads; stacks material at right angle from any 6-ft aisle. Offset drive wheel has 200-degree turning arc. Forks extend 24 in. and back in a few seconds' time. Handles any size pallets without any changes being





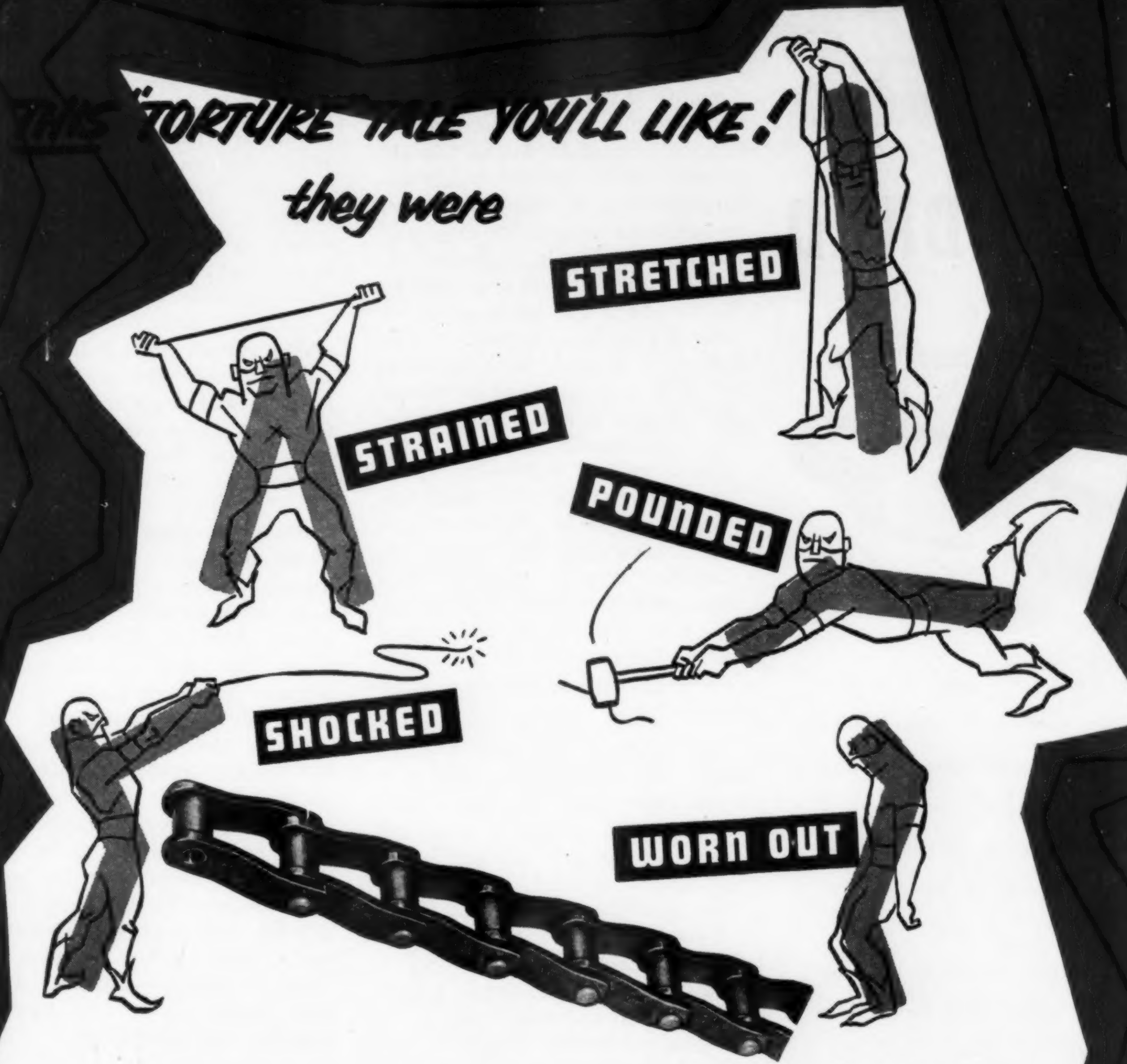
made in the truck. Pallets can be stacked close together, since no space is needed between loads to accommodate base forks. Lightweight; may be operated safely on low-capacity elevators and floors. Has 51-in. free lift, making possible the elevation of loads in low-ceiling areas, boxcars and truck trailers without increasing truck's overall height. *Raymond Corp., Greene, N. Y.*

Electric Lift Trucks: Standard Skylift CF line features class "H" silicone insulated motors, duolift rams with leakage return, high lift speed, hydraulic safety fuses and easy accessibility for maintenance. Have upright channels of heavy rolled steel, interchangeable carriage and upright rollers mounted on sealed ball bearings, and widely spaced adjustable side thrust rollers on fork carriage. Models available with rated capacities of 1000, 1500, 2000 and 3000 lb with 48-inch load length and 4000 lb with 30-inch load length. All models offered with either duolift rams, providing full free lift feature, or monolift, permitting 19¼ in. free lift before overall height begins to increase. All models have full telescopic lift height of 132 in. Have four speeds in both forward and reverse. Truck can be reversed only in first speed, and brake is interconnected with seat, so that truck is automatically braked when operator leaves seat. *Automatic Transportation Co., Chicago, Ill.*

Metalworking

Profile Mill: For precision milling of profiles, irregular shapes, hexagons and geometric forms. Will hold dimensional relations to a single dimension tolerance. Only two milling operations—rough and finish—are required. Cams and fixtures are built to specifications. Machine can be set up for various jobs by changing cams. *Spike Mfg. Co., Farmington, Mich.*

Automatic Lathe: Designed to operate at high speeds, 60-in., heavy-duty contour turning unit uses tungsten-carbide tools; machines steel mill rolls. Electronically controlled cross-feed and lon-



...and subjected to every conceivable test that a modern engineering "torture chamber" could devise. From it came a great new line of improved Rex Chabelco Steel Drive and Conveyor Chains. It wasn't easy because, as Chabelco users know, the previous Chabelco Chains were "tough," strong...hard to beat. But by making a few important design and manufacturing changes, Rex engineers produced a line of improved standard drive and conveyor chains. We call them Rex Chabelco "R" Series Drive and "RR" Series Conveyor Chains! You'll call them the best chains you've ever used!

Why not have your Rex Field Sales Engineer give you the complete story. Chain Belt Company, 4643 W. Greenfield Avenue, Milwaukee 1, Wisconsin.

CHAIN BELT REX COMPANY

Chain Belt COMPANY OF MILWAUKEE

® Atlanta • Baltimore • Birmingham • Boston • Buffalo • Chicago • Cincinnati • Cleveland • Dallas Denver • Detroit • El Paso • Houston • Indianapolis • Jacksonville • Kansas City • Los Angeles • Louisville Midland, Texas • Milwaukee • Minneapolis • New York • Philadelphia • Pittsburgh • Portland, Ore. West Springfield, Mass. • St. Louis • Salt Lake City • San Francisco • Seattle • Tulsa • Worcester

Distributors located in principal cities in the United States and abroad
EXPORT OFFICES: Milwaukee and 19 Rector Street, New York City

gitudinal feed motors are actuated by signals from a floating stylus mounted on the carriage cross slide to produce exact contour duplication of templates. Lathe has all-geared, totally enclosed headstock, three-way 65-in.-wide bed, two steady rests or roll housings and live-center tailstock. Contours roll bodies while rolls are supported on their necks in roll housings rather than on centers. Unit will turn on centers if desired. Roll housings are fitted with bearings which adjust radially to suit neck diameter and longitudinally to contact thrust faces at ends of roll body. Housings accommodate neck diameters from 11 to 28 in. Tailstock is a special housing fitted with an oversize steel spindle operated by a worm and worm wheel. Carriage has extra long wings which bear on the bed for their entire length. *Mackintosh - Hemphill Co., Pittsburgh, Pa.*

Die Sinking Machine: Model KF12 universal pantograph performs copy milling operations ranging from die sinking of large and intricately shaped dies and molds to light engraving. Reproduces dies and molds accurately and economically with high surface finish. Pantograph system permits two or three-dimensional milling at 1:1 ratio and reductions or enlargements from 1:1.5 to 1:4. Cutting tool covers an area of 15¾ by 15¾ in. or 10 by 19¾ in. Cutting areas can be fully utilized with die blocks up to 15¾ in. high. Power-operated vertical saddle permits raising both master and workpiece simultaneously at a rate of 30 in. per minute, which reduces downtime for setting up and tool changes. Rough milling spindle has built-in speed reducer. Infinitely variable spindle speeds range from 60 to 10,000 rpm. Size of machine, 54 in. long, 67 in. wide, 77 in. high; weight about 3200 lb. *Cosa Corp., New York, N. Y.*

Production Drill Press: For multiple operations involving drilling, tapping, reaming, burring and chamfering. Four types, equipped with from one to four drill heads, available. Each drill head is a 14-in. drill press with a simple belt tension release which provides quick speed changes. Free-floating spindle eliminates side thrust and whip; sealed ball bearings simplify maintenance. Spindle speed range is approximately 720 to 4325 rpm. Maximum capacity, 7 in. from spindle to column and 18 in. from chuck to table. Drill head positioning attachment gives each drill head assembly 4 in. of vertical adjustment at any one setting. Attachment can be set at any point on the column and after being locked in place, swivels around the column with the drill head and moves the head up and down the column by means of an enclosed worm gearing. *South Bend Lathe Works, South Bend, Ind.*

Rotary Dividing Table: "Wafer-Thin" model built by Hayes Engineers Ltd., Leeds, England, is thinnest rotary dividing table manufactured. Provides highest capacity under the tool on all milling, drilling and jig boring machines. High precision unit; maximum error through 180 degrees is 0.00025. Hardened steel worm is fitted into an eccentric quill assembly so that it can be disengaged. Dead stop is fitted to allow worm to be re-engaged to correct position. Has large micrometer dial as well as a divided ring around periphery of table. Movable zero datum is provided to facilitate reading. Available in either 11 or 12-in. diameter. *International Machinery Div., British Industries Corp., New York, N. Y.*

Jig Borer: No. 2E for precision boring and milling. Features Electrolimit measuring system and preloaded ball roll quill. Employs separate measuring units for longitudinal and transverse movements. Both units are independent of traversing screws; each obtains basic 1-in. spacings electromagnetically and registers a zero reading on an indicating meter without making physical contact. Micrometer head is graduated for reading to 0.0001-in. Ball roll quill "roll-feeds" on 288 precision balls preloaded between hardened quill and liner with a total bearing pressure of 6000 lb. Stability of ball roll quill makes milling feeds practical for the table and carriage. Automatic interlocking mechanism clamps head posi-

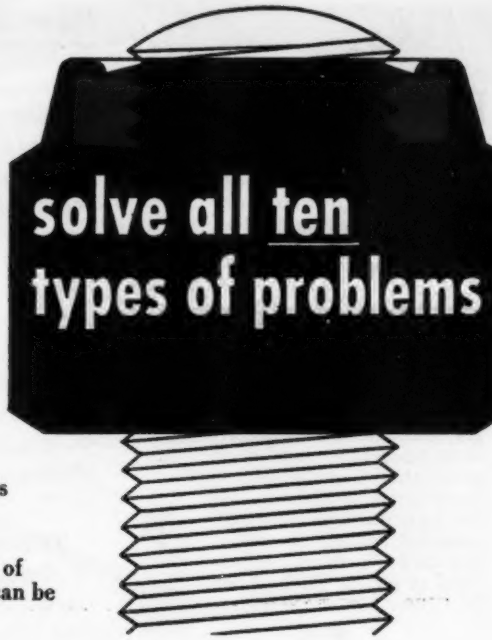
tively in position on column after motion and releases it before motion starts. Electronically controlled milling feeds are infinitely variable from 1 to 15 in. per minute. Eight spindle feeds range from 0.0005 to 0.015 in. per revolution, both up and down. Twelve spindle speeds ranging from 37 to 1800 rpm. Table is 22 by 44 in.; travel is 36 in. longitudinally and 22 in. transversely; maximum height between table and spindle end is 27 in. *Pratt & Whitney, Div. Niles-Bement Pond Co., West Hartford, Conn.*

Plate - Sheet Working Machine: Developed by GV Nibbler Co., Stockholm, Sweden; available in three sizes. Performs beading, straight cutting, folding, joggling, circular cutting, slot cutting, nibbling, irregular or free-hand cutting and louver cutting operations. Mechanism is enclosed and operates in an oil bath. Cuts mild steel, stainless steel, aluminum, brass, copper, fiber, etc. Cutting capacity of Junior model, 1/8-in.; Medium model, 15/64-in.; Senior model, 5/16-in. Throat depths are 28½, 40 and 49 in., respectively. All models have 220 or 440-v, 60-cycle, three-phase motors. Circle, straight cutting, folding and beading tools as well as adjusting gages and necessary spanners are included; other tools are available for special operations. *Eric S. Johnson Co., Chicago, Ill.*

Boring and Facing Machine: For irregular profile milling and boring of aluminum and magnesium castings and forgings or cast and rolled armor plate. Has duplicator equipment with templates. Especially applicable to profile machining necessary in the manufacture of aircraft and armored vehicle parts. Features a motor-powered spindle housing unit that moves directly into work face. *Forney's Inc., New Castle, Pa.*

Wet Carbide Grinder: Model AW for various rough and finish jobs. Provides ample flow of water for adequate cooling when a considerable amount of carbide must be removed. Features quick-acting indexing tables which can be set instantly to required angle and do

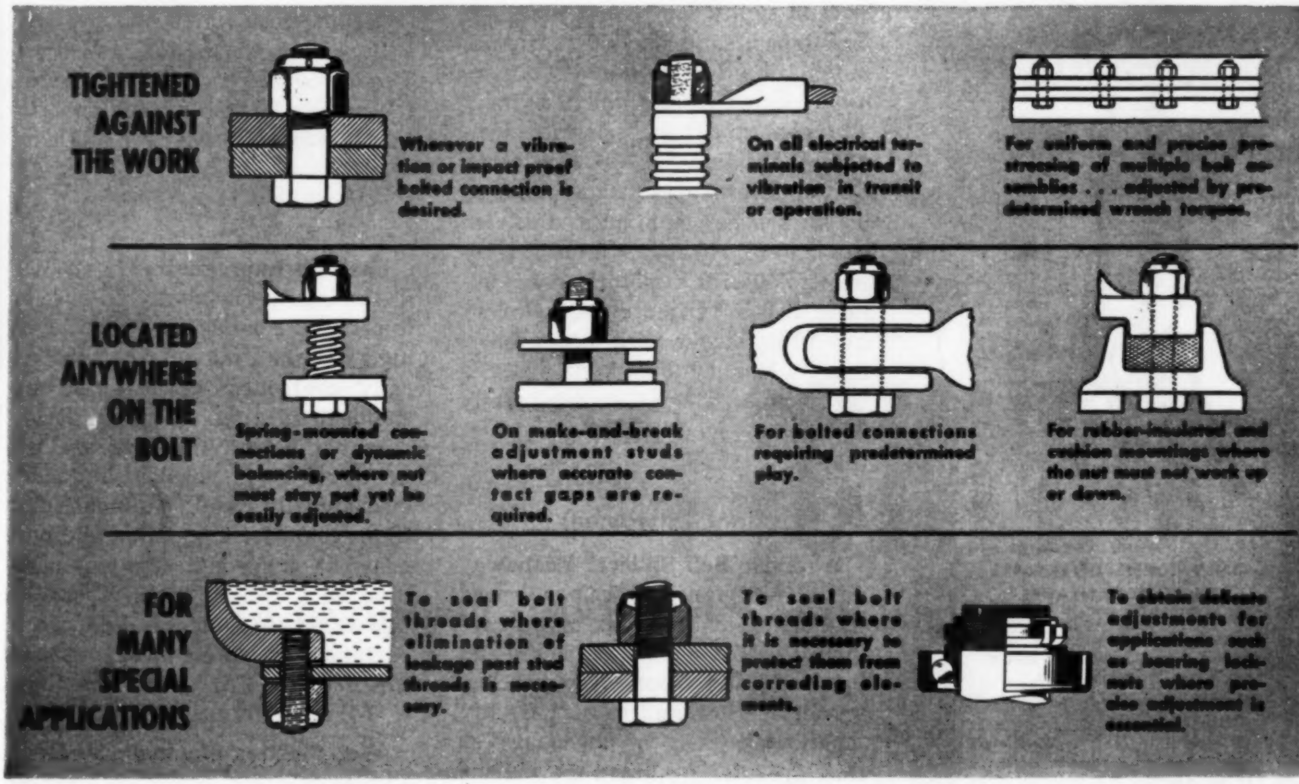
elastic stop nuts



Here are ten typical fastening problems. One device, the ELASTIC STOP nut, solves them all—without additional parts or operations. Deliberately undersized in relation to bolt diameter, the red elastic collar grips the bolt with a perfect fit, exerting a continuing self-locking pressure against the threads, and holding the nut securely in place at any point on the bolt. It also provides a tight seal against the bolt threads, which prevents seepage and wear-producing axial play. And because the bolt threads are protected against moisture from without, the nuts are not "frozen" to the bolt by corrosion.

ELASTIC STOP nuts stay tight, right where you put them, in spite of vibration and stress reversals. Yet they are not jammed in place, and can be removed with a wrench and reused many times.

For further information on ESNA self-locking fasteners, mail the coupon below.



ELASTIC STOP NUT CORPORATION OF AMERICA

ESNA TRADE MARK

ROLLPIN TRADE MARK

Dept. N41-14, Elastic Stop Nut Corporation of America
2330 Vauxhall Road, Union, New Jersey

Please send the following free fastening information

☐ Elastic Stop nut bulletin ☐ Here is a drawing of our product. What self-locking fastener would you suggest?

☐ Rollpin bulletin

Name________________ Title________

Firm________________

Street________________

City__________ Zone____ State____

New Machines

not require locking screws or other devices to keep them in position. Has separate motor-driven coolant pump, pan, settling tank, piping, valves, nozzles and spray guards. Also has drum type on-off-reverse switch, combination protractor tool guide and truing holder. Furnished with ½-hp motor for 110-v, 60-cycle single phase or 220/440-v, 60-cycle, three-phase current. Overall dimensions, 22 by 29 by 50 in.; weight, approximately 365 lb. *Thomas Prosser & Son, New York, N. Y.*

Trimming Press: Model No. 210 is designed for rigidity and quiet operation and features streamlined box type crown construction. Single-geared unit has twin herringbone drive gears running in oil, rim type oil-tight gear guards, air counterbalance concealed in uprights, automatic lubrication and cool running friction clutch. Friction clamp slip type knockout prevents accidental breakage of dies or other parts. Press operates at a speed of 35 strokes per minute with ram capacity of 440 tons. Slide has 16-in. stroke with a motorized adjustment of 6 in.; trimming attachment has 10-in. stroke with 4-in. adjustment. Has 86 by 94-in. base, is 18 ft high, weighs approximately 98,000 lb. *E. W. Bliss Co., Canton, O.*

Plant Equipment

Abrasive Belt Splicer: Econaway model, redesigned to increase speed of splicing and also improve splice quality. Motor is now enclosed and belt drive and arbor carry grinding wheel. Cutter and press bars are mounted on top of enclosure so that operator need not move from one station to another. Electrical curing press reduces curing time from 24 hours to less than 60 seconds. Saves space formerly required for storage of belts during curing. Machine can splice belts up to 14 in. wide. *Aget-Detroit Co., Ann Arbor, Mich.*

Coolant Clarifier: Features over 16,000 sq in. of filtering area. Removes all chips, abrasives, dirt and other solid contaminants from both water-based and mineral oil coolants used in individual machine tools or small central systems. Liquid passes through multiple filtering tubes suspended vertically within clarifier. Tubes are self-cleaning. Unit can be operated continuously, independent of machine tool, or intermittently. *Honan-Crane Corp., Lebanon, Ind.*

Processing Equipment

Cabinet Ovens: Equipped with high-pressure motor-driven blower which propels heated air in a definite air-flow pattern through work chamber, preventing disturbances due to radiant heat and assuring temperature uniformity. Has indicating temperature control, high-volume adjustable air flow, high and low heat switch for close control and quick recovery, manual interlock for purge period operation of blower without heat, electrical interlock for turning off heat in case of blower motor failure, adjustable positive exhaust and intake. Available in either horizontal or vertical flow. Provide temperatures to 650 and 850 F for 220 or 440-v operation. *Grieve-Hendry Co. Inc., Chicago, Ill.*

Castings Impregnator: Seals metal castings permanently. Self-contained unit comprised of autoclaves, high-vacuum pump, thermostatically controlled wash tanks, mechanical surging unit and power-operated hoist. Castings are run in batch loads; basket is loaded mechanically and automatically. Impregnant level, overflow and pore penetration are indicated and measured on a cycling panel. Occupies less than 40 sq ft of floor space, exclusive of baking oven. *American Metaseal Mfg. Corp., New York, N. Y.*

Wet Blasting Machine: Model 30 Liquamatte for precision cleaning and finishing applications involving small pieces that can be lifted and handled manually. Recommended for small stamping, die-casting and drawing dies; glass, plastic and small rubber molds; cleaning oxides from tungsten carbide tips before brazing to cutting tools; removing heat treat scale; deburring. Fine mesh abrasives suspended in water are propelled at the work by compressed air. Flat surfaces, knurling, and other vulnerable areas are not damaged

New Machines

by abrasives. Tolerances of 0.0001-in. can be maintained where required. Features pushbutton controls, vertical pump for slurry agitation and recirculation. Slurry is not lost through leakage because there are no packing glands. Hopper need not be drained before pump is removed. Size, 30 by 30 in. at base, 83 in. high; blasting compartment is 30 in. square and 31 in. high. *American Wheelabrator & Equipment Corp., Mishawaka, Ind.*

Testing and Inspection

Monochromatic Colorimeter: Determines color densities and percentage of light transmission in quantitative analyses. Employs transmission type interference filters which provide high spectral purity; detects small absorption bands in transmission curve. Filters will not deteriorate with age because they utilize metallic films evaporated on glass surfaces. Features two-knob control. Dual-purpose filter holder accommodates 2 by 2-in. or 1 by 1-in. filters. Light is automatically cut off from photoelectric cell when holder is withdrawn from light path. Scale gives direct transmission and density readings by means of a floating spotlight. *Bausch & Lomb Optical Co., Rochester, N. Y.*

Dynamometer: Tru-Tork Model 52 Custom for testing ac and dc motors. Features self-regulating absorption unit and "Quick Check" torque scale which indicates directly in foot-pounds and horsepower. Self-regulating brake assures constant torque loading regardless of speed variations of motor under test, making possible the simulation of conditions similar to those encountered in actual motor operation. Tests can be made by unskilled personnel without the use of computations or formulas. Switch-gear offers selection of either single or three-phase power. Meters indicate ac volts, ac amperes, ac watts and speed of the motor while in operation. Rated at 3 hp at 1800 rpm; has speed range extending to 4000 rpm. *George L. Nankervis Co., Detroit, Mich.*

MEN OF MACHINES

John T. Davidson has been elected to the new post of vice president of the Standard Register Co., Dayton, O., in charge of engineering and research. Educated at Ohio State University, Mr. Davidson was associated with the National Cash Register Co. in Dayton prior to joining Standard Register in 1940. Until his new appointment he held the position of chief engineer. A primary function of the division which Mr. Davidson now heads is fitting automatic forms operation to successive new advancements in business machines through mechanical, electrical and electronic devices which the company makes for continuous, controlled form feed and for auxiliary processing of the documents. He will direct a staff of over 100 engineers.

John T. Davidson

♦

Gordon B. Carson, former engineering faculty member at Case Institute of Technology, was named dean of the College of Engineering at Ohio State University. Mr. Carson, who was secretary of the Selby Shoe Co., as well as manager of engineering, succeeds the late **Charles E. MacQuigg.** Mr. Carson received a degree in mechanical engineering from Case in 1931 and became a graduate assistant at Yale University, serving until 1932, when he received his M.S. in mechanical engineering. He returned to Case

Gordon B. Carson

to teach and was promoted to assistant professor and then to associate professor of industrial engineering in charge of the industrial division of mechanical engineering. In 1938 he was awarded the professional degree, "Mechanical Engineer," by Yale University. While at Case, Mr. Carson served as research engineer and then as director of research for the Cleveland Automatic Machine Co. He also developed the curriculum for and initiated the graduate program leading to the degree of M.S. in industrial engineering at Case. In 1944 he left the school to become assistant to the general manager of the Selby Shoe Co., devoting his attention to special management projects. After a year he became manager of engineering and in 1949 was also made secretary of the corporation. Mr. Carson is a member of Tau Beta Pi and Sigma Xi, the American Society of Mechanical Engineers, the National Society of Professional Engineers, the Industrial Management Society and the American Ordnance Association.

♦

Aro Equipment Corp., Bryan, O., has appointed **Arthur S. Iberall** to direct the newly formed research and development department in its Aircraft Div. in Cleveland. In addition, Mr. Iberall will serve as a consultant to Aro customers desiring technical guidance.

♦

The Babcock & Wilcox Co., New York, recently announced several changes in engineering personnel. **W. H. Rowand,** chief engineer of the Boiler Div., has been elected a vice president of the company. He will continue to hold the post of divisional chief engineer. **Paul R. Loughin,** former chief of staff engineering and recently named an executive assistant, has been transferred to the staff of the Boiler Div. **C. L. Norton Jr.,** former technical director, was named executive assistant in charge of all development and engineering activities in the company's Refractories Div. His former assistant, **Bradford Hooper,** was named technical director.

♦

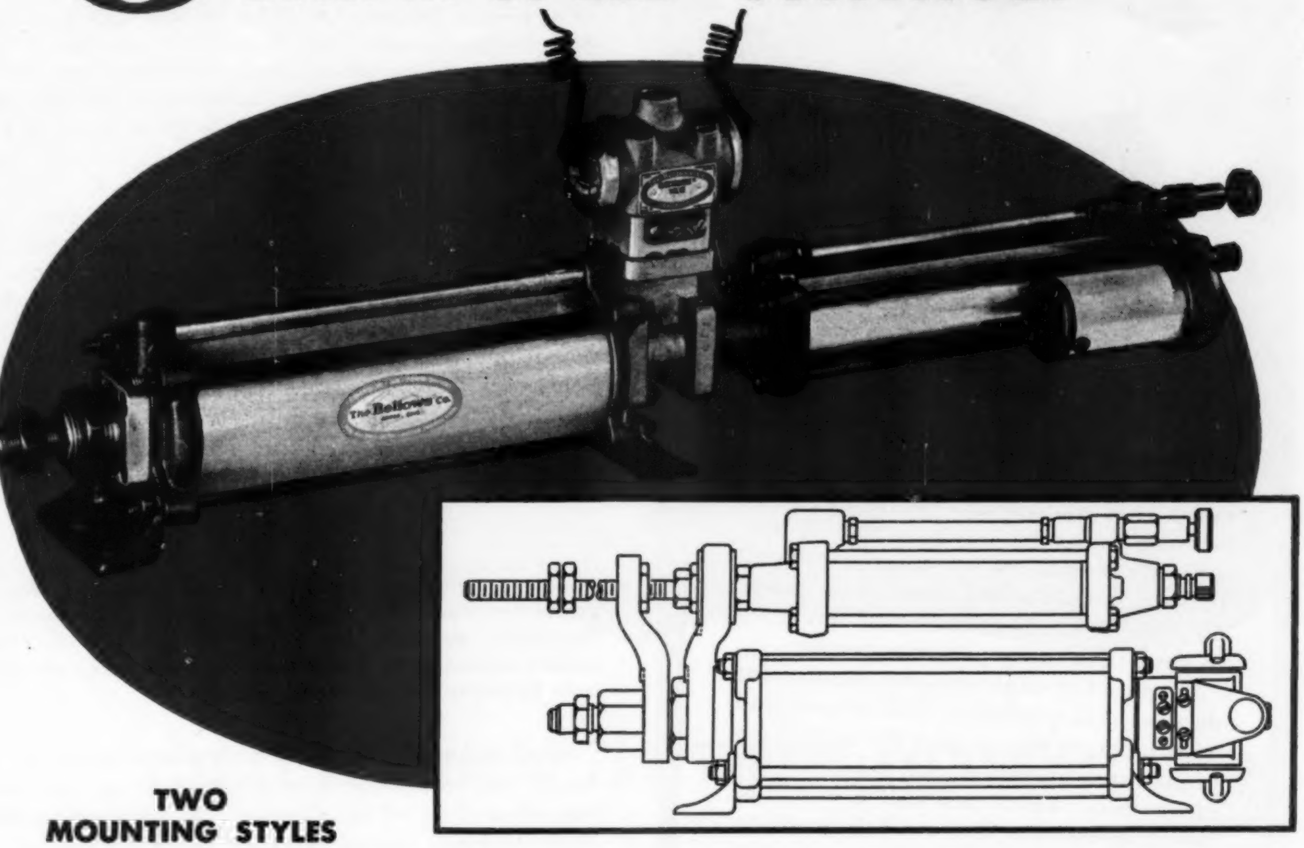
Formerly chief engineer, **Homer Gray** has been named executive vice president of Air-O-Matic Power Steer Corp., Cleveland.

♦

Officers of the American Gear Manufacturers Association for 1953-1954 elected at the recent annual meeting are as follows: president, **George H. Sanborn,** sales manager, Fellows Gear Shaper Co.; vice president, **R. B. Holmes,** general manager, Link-Belt Co.; treasurer, **Fred R. Eberhardt,** president, Eberhardt-Denver Co. The association's executive committee for the coming year will be composed of **R. B. Tripp,** executive vice president, Ohio Forge and Machine Corp.; **R. E. Smallwood,** engineer, industrial division, Dominion Engineering Works; **E. C. Denne,** manager, gear department and consultant, United Engineering and Foundry Co.; and **T. A. Jones,** president, W. A. Jones Foundry and Machine Co.

This Air Cylinder Combines

1 BUILT-IN HYDRAULIC CONTROL
2 BUILT-IN ELECTRIC CONTROL
3 BUILT-IN SPEED CONTROLS



TWO MOUNTING STYLES

The Bellows Air Motor and Hydro-Check (Model HCB-EM) is available arranged in tandem as shown above, or in parallel, as sketched.

●

Combination units of Air Motor and Hydro-Check are also made with any one of six other Bellows directional valve arrangements; and, of course, Air Motors and Hydro-Checks are available as separate units.

In one compact packaged power unit are built-in controls to cover every phase of air cylinder operation. Built-in hydraulic control of piston rod movement gives the characteristic smoothness of hydraulic operation, yet retains the speed and flexibility of air power. Built-in electrically controlled directional valve permits quick, positive, remote control, or easy electrical synchronization with other machine elements. Built-in speed controls permit exact control of piston rod speed in either or both directions.

This combination of Bellows Air Motor and Hydro-Check is available in five bore sizes: 1¼", 1¾", 2½", 3⅝" and 4½" in standard stroke lengths up to 18".

Here is air cylinder power at its best — fast, flexible, safe, and smooth.

BULLETIN CL-50 TELLS THE STORY. WRITE FOR IT TODAY!
DEPT. MD753

Controlled-Air-Power

The Bellows Co.

AKRON 9, OHIO

699-A

FOR FASTER, SAFER, BETTER PRODUCTION

THE ENGINEER'S

Library

Recent Books . . .

Engineering Drawing. *By T. E. French, late professor of engineering drawing, and C. J. Vierck, professor of engineering drawing, Ohio State University; 725 pages, 6½ by 9½ inches, clothbound, published by McGraw-Hill Book Co. Inc., New York; available from* MACHINE DESIGN, *$8.00 postpaid.*

A classic in its field since its first publication in 1911, this eighth edition of *Engineering Drawing* has been revised to follow the latest standards and incorporate all the modern methods and practices. The chapters have been rearranged into four basic divisions: fundamentals of shape description, size descriptions, basic machine elements, and a section of working drawings. Many new drawings, sketches, and illustrations have been added along with practically a new chapter on orthographic reading.

♦

Direct Current Machines for Control Systems. *By A. Tustin, professor of electrical engineering, Birmingham University, England; 320 pages, 6 by 9 inches, clothbound; published by MacMillan Co., New York; available from* MACHINE DESIGN, *$10.00 postpaid.*

Application of automatic precision controls in industry has been materially advanced by the introduction of a class of dc dynamo, the rotary amplifier. Such machines, known variously as metadynes, amplidynes, and many other names, provide power amplification with a fast speed of response and thus furnish a basis for new solutions to control problems.

Details of servo design and performance are fitted together with details of machine design for power amplification. An explanation of the principles on which these machines operate provide a basis for their further development, both in design and application.

♦

Mechanical Engineering Thermodynamics. *By David A. Mooney, mechanical engineer; 552 pages, 6 by 9 inches, clothbound; published by Prentice-Hall Inc., New York; available from* MACHINE DESIGN, *$9.35 postpaid.*

Thermodynamics is treated in this textbook as a logical formulation of facts known from practical experience. The general order of presentation is basic concepts, principles, data on substances, and engineering applications. The basic concepts of work and heat are discussed in detail along with a number of illustrative examples. The properties of substances are presented from a general viewpoint and then specific

items are considered. In the chapters on application, care is taken to point out the considerations other than efficiency which influence the use of a particular cycle or apparatus.

The appendix is made up of a nomenclature and eleven tables. Large-size temperature-entropy diagrams for carbon dioxide and air are included in a back-cover pocket along with the Molier steam chart, a pressure-enthalpy chart for Freon-12, and psychrometric chart.

•

Mechanism. *By Joseph Stiles Beggs, associate professor of engineering, University of California, Los Angeles;* 200 *pages,* 8½ *by* 11 *inches, loose-leaf; available from U. C. L. A. Students Store,* 402 *Westwood Blvd., Los Angeles* 24, *Calif.;* $3.00.

Written from notes for an advanced course in kinematics, this volume provides a reference for the selection of mechanisms. Mathematical and graphical analyses are presented concerning position, velocity, acceleration, rolling curves, gearing, cams, couplings, intermittent drives, linkages and slides, tension and compression links, mechanisms trains, computing mechanisms, and controlling circuits.

New Standards

Involute Spline and Serration Gages and Gaging. *ASA B*5.31-1953; 22 *pages,* 8½ *by* 11 *inches, paperbound; available from American Society of Mechanical Engineers,* 29 *W.* 39*th St., New York* 18, *N. Y.;* $1.00 *per copy.*

Factors which affect fits and their control, working and inspection gages, inspection methods, gage numbering and identification, machining tolerances and allowable errors are among the subjects covered in this involute spline gaging standard. Material also included are gage dimensioning and design, ring gage pin measurements, comparison of serrations with splines, serration gage tolerances, and formulas for serration plug and ring gages.

•

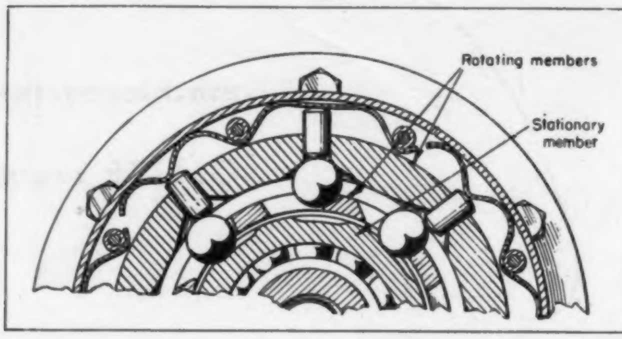
Code for Pressure Piping. *ASA B*31.1*a*-1953; 24 *pages,* 8½ *by* 11 *inches, paperbound; available from American Society of Mechanical Engineers,* 29 *W.* 39*th St., New York* 18, *N. Y.;* $1.00 *per copy.*

This supplement lists changes and additions in the formulas, tables, and text of the ASA B31.1-1951 standard. Tables of allowable pressure values for pipe of various materials used in district heating, power, and oil piping systems are included. Two tables give pressure-temperature ratings of American standard cast and forged, 5 per cent chrome-molybdenum alloy steel valves, fittings, and flanges when used with ring and other joints.

NOTEWORTHY

Patents

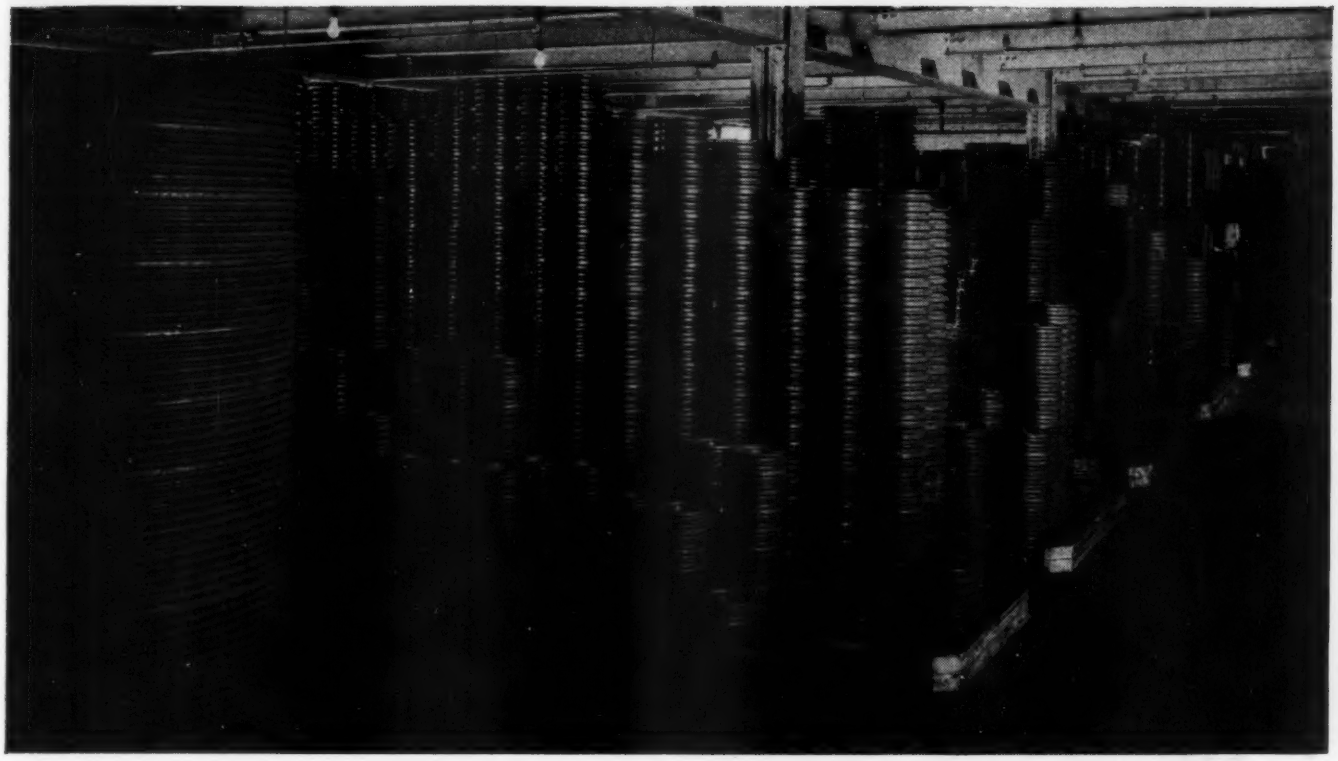
DIRECTIONAL TORQUE CONTROL for drive connections is afforded by the mechanical shaft coupling detailed in patent 2,626,027. Designed by Roy H. Anderson, the coupling can be operated in either direction of rotation and incorporates an automatic locking action which prevents torque reversals under load. Power is transmitted to the outer driven member by two sets of steel balls spaced alternately in wedge-shaped slots in the driving member which rotates about a third stationary member. Direction of the applied torque determines which set of balls is in driving contact. In normal operation the balls ride over the stationary member; however, tendency of the driven member to reverse torque direction under load causes a wedging action to lock the coupling against rotation. Oscillation of the balls during rotation is prevented by spring-actuated detents. Assignee of the patent is Boeing Airplane Co.



DUAL SCREW DRIVE for transmitting linear motion at high speed offers two-unit reliability in a single mounted assembly. Covered in patent 2,630,022 assigned to Boeing Airplane Co. by Frank Terdina, a ball-bearing screw and nut drive transmits power to a nonrotating housing mounted to the nut. Two worm gear units power the screw and nut independently, driving the housing at high speed when both are operating simultaneously. Use of two power units assures continued operation if one fails. Design of the drive permits reversal only under power.

DROPTIGHT SEALING of journal bearings is obtained with the seal shown in patent 2,626,839 without the necessity for close fits usually required with conventional types. Assigned to Ross Gear & Tool Co.

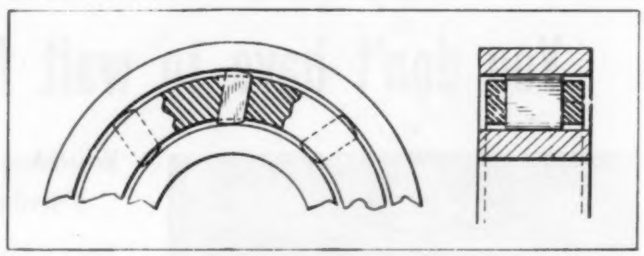
(*Concluded on Page 312*)



(Concluded from Page 308)

by William K. Creson and Matthew W. Berghoff, the V-shaped all-rubber or synthetic seal fits into a counterbored recess in the journal end. One lip checks escape of lubricant along the shaft and the other prevents losses over the seal outer surface. Takeup adjustment for diameter variations in assembly is provided by serrations in the outer surface. Seal retention requires only a minimum of axial force and may be accomplished by any of several methods such as snap or retaining rings, staking, etc.

UNIFORM CLUTCHING ENGAGEMENT is achieved with floating sprags in an overrunning clutch designed by David E. Gamble. Assigned to Borg-Warner Corp. under patent 2,624,436, the design employs a rubber ring carrying the sprags in axial slots, offset to provide clutching action. Resilient action of



the ring permits slight movement of the sprags, assuring proper seating and uniform pressure of engagement. In addition, the construction simplifies assembly and facilitates manufacture of clutches in small sizes. A modification providing secure retention of the sprags in the carrier is also shown.

TORSIONAL VIBRATION DAMPING of connected shafts is afforded by a flexible coupling which can also be operated under angular misalignment. Described in patent 2,620,640, the coupling has been designed for use on internal combustion engine crankshafts operating under temperatures up to 180 F. Flexibility is provided by a floating sleeve member which transmits power from the driving to the driven shaft connections through a vibration-absorbing rubber cushioning element—operation is essentially that of an Oldham coupling. Construction of the coupling, which limits axial separation of the members, is described in several modifications. The patent has been assigned to General Motors Corp. by Max G. Bales.

Copies of all patents presented in this department may be obtained for 25 cents each from The Commissioner of Patents, Washington 25, D. C.

FOR GREATER SAFETY

Greyhound Specified VICKERS HYDRAULIC POWER STEERING

on 400 New Coaches

Always searching for ways to further increase passenger safety, Greyhound Lines selected Vickers Hydraulic Power Steering for the 400 new Model PD-4104 GM Coaches recently purchased.

Vickers Hydraulic Power Steering prevents the possible loss of driver control when a vehicle is forced off the pavement onto a soft shoulder . . . or when a front tire blows out. Extra steering power and quick maneuverability are always available for emergency conditions.

The Vickers System absorbs all road shock and transmits it to the vehicle frame . . . there can be no kick-back at the steering wheel. The driver supplies only enough effort to slightly move a servo valve . . . fatigue is thus greatly reduced and the driver is more alert.

Only minor alterations are usually required to incorporate Vickers Hydraulic Power Steering in new and existing vehicle designs. For further information, ask for new Bulletin M-5104.

VICKERS Incorporated
DIVISION OF THE SPERRY CORPORATION
1430 OAKMAN BLVD. • DETROIT 32, MICH.
Application Engineering Offices: ATLANTA • CHICAGO (Metropolitan) CINCINNATI • CLEVELAND • DETROIT • HOUSTON • LOS ANGELES (Metropolitan) NEW YORK (Metropolitan) • PHILADELPHIA • PITTSBURGH • ROCHESTER ROCKFORD • SEATTLE • TULSA • WASHINGTON • WORCESTER

VICKERS *hydraulic*
POWER STEERING
is Effortless
Positive and Shockless

Vickers Hydraulic Power Steering Booster does the actual work of steering

Engine-driven Vickers Vane Pump (with integral volume control and relief valves, and oil reservoir) supplies power for steering.

6231

ENGINEERS AND BUILDERS OF OIL HYDRAULIC EQUIPMENT SINCE 1921